STEAM BOILERS

TERRELL CROFT, Editor

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PREFACE TO THE SECOND EDITION

Since Terrell Croft prepared the original text of this book in 1922, many changes have taken place in boiler-room practice —new equipment has been developed and much of that shown in the first edition has become obsolete. As Mr. Croft was unable to undertake the revision, the writer was authorized to do so.

I have endeavored to maintain the author's original style of presentation and to write for the "men of little schooling" indicated in the author's preface.

The division on Modern Types of Boilers has been entirely rewritten and divided into two parts—one division on Fire-tube Boilers and one on Water-tube Boilers. In these divisions material has been added on steam driers, steam washers, waterwalls, and superheaters. The divisions on boiler materials, stresses, and joints have been made to agree with the 1933 A.S.M.E. Boiler Construction Code (the latest available when this work was undertaken) and material added on welding. New text and illustrations have been prepared on stokers, pulverized coal, oil burning, boiler settings, furnaces, economizers, air preheaters, fans for induced and forced draft, and feed-water treatment. With these additions and other changes throughout the text, it is felt the book now gives an adequate picture of present-day boiler-room practice.

Most of the illustrations of boilers, stokers, and oil burners have been taken from *Power*. Other illustration material has been obtained through the cooperation of many manufacturers, and credit for these pictures has been given throughout the book.

R. B. PURDY.

NEW YORK CITY,
February, 1937.

PREFACE TO THE FIRST EDITION

"Steam Boilers" has been prepared primarily for men of little schooling who desire to acquaint themselves regarding this subject. A working knowledge of arithmetic will qualify one to read it understandingly. But, since all of its statements and the methods and principles which it proposes are both theoretically and practically sound, it may be used effectively by any one, regardless of his training or experience, who seeks steam-boiler information. It has been written with the special intent of serving the needs of those who are preparing to pass engineers'-license examinations.

Drawings for all of the 514 illustrations were made especially for this work. It has been the endeavor to so design and render these pictures that they will convey the desired information with a minimum of supplementary discussion.

Throughout the text, principles which are presented are explained with descriptive expositions or with worked-out arithmetical examples. At the end of each of the 25 divisions there are questions to be answered and, where justified, problems to be solved by the reader. These questions and problems are based on the text mater in the division just preceding. If the reader can answer the questions and solve the problems, he then must be conversant with the subject matter of the division. Detail solutions to all of the problems are printed in an appendix in the back of the book.

As to the general method of treatment: First, the functions, the history and the modern types of boilers are considered. Then boiler codes and inspection laws are discussed. Next, the elements of modern boiler construction, in accordance with The American Society of Mechanical Engineer's Boiler Code, is presented under such division titles as: Boiler stresses and strengths, riveted joints, braces and stays, fire tubes and water tubes, manholes and handholes—and the like.

This is followed by matter relating to: boiler accessories, steam generation and superheating, and boiler capacities and ratings.

Continuing: the matter which concerns boiler-room economy is introduced under such headings as: Fuels, draft and its production and measurement, combustion and firing, boiler settings and furnaces, mechanical stokers, petroleum and gaseous fuels, chimneys breechings and dampers, artificial draft equipment, fuel economizers, feed water and feed-water treatment—and steam-boiler management inspection and maintenance.

Finally, the selection of steam boilers is given attention.

With this, as with the other books which have been prepared by the author, it is the sincere desire to render it of maximum usefulness to the reader. It is the intention to improve the book each time it is revised and to enlarge it as conditions may demand. If these things are to be accomplished most effectively, it is essential that the readers cooperate with us. This they may do by advising the author of alterations which they feel it would be desirable to make. Future revisions and additions will, insofar as is feasible, be based on such suggestions and criticisms from the readers.

Although the proofs have been read and checked very carefully by a number of persons, it is possible that some undiscovered errors may remain. Readers will confer a decided favor in advising the author of any such.

Terrell Croft.

University City,
St. Louis, Mo.,
May, 1921.

ACKNOWLEDGMENTS

The author desires to acknowledge the assistance which has been rendered by a number of concerns and individuals in the preparation of this book.

Considerable of the text material appeared originally as articles in certain trade and technical periodicals among which are: *Power*, *National Engineer*, *Power Plant Engineering*, *Southern Engineer*, and *Combustion.*

Among the concerns which cooperated in supplying text data and copy for illustrations are: Freeman & Sons Manufacturing Company, boiler manufacturers; Murray Iron Works Company; Babcock & Wilcox Company; Heine Boiler Company; Westinghouse Machine Company; Riley Stoker Company; Taylor Stoker Company; Stirling Boiler Company; Page Burton Boiler Company; Riehle Testing-Machine Company; Ashton Valve Company; Crosby Steam Gage & Valve Company; Penberthy Injector Company; Ohio Brass Company; Precision Instrument Company; Kewanee Boiler Company; American Blower Company; B. F. Sturtevant Company; Green Fuel Economizer Company; Edgemoor Boiler Company; Smooth-On Manufacturing Company.

Special acknowledgment is accorded to the American Society of Mechanical Engineers for permission to incorporate herein certain tables and other material from the A. S. M. E. Boiler Code. Also, acknowledgment is accorded to The Hartford Boiler Insurance Company for the use of data from certain of its publications.

Certain material in the text has also been taken from the following sources: "Steam—Its Generation and Use, by the Babcock & Wilcox Company; *University of Illinois Bulletin* 31, "Fuel Economy in the Operation of Hand-fired Power Plants." Engine-cylinder drawings in the division on Stresses and Strains were taken from articles by F. R. Low in *Power*. The

chronology of power-plant apparatus is due to Chas. J. Mason in *National Engineer*, April, 1914.

Other acknowledgments have been made throughout the book. If any has been omitted, it has been through oversight and if brought to the author's attention it will be incorporated in the next edition.

TERRELL CROFT.

CONTENTS

LIST OF SYMBOLS

The following list comprises practically all of the symbols which are used in formulas in this book. Symbols which are not given in this list are defined in the text where they are first used. When any symbol is used with a meaning different from that specified below, the correct meaning is stated in the text where the symbol occurs.

Symbol	Meaning	Section first used
A	Area, in square inches	238
A	Cross-sectional flue area of chimney, in square feet	518
A_d	Cross-sectional area of diagonal stay, in square inches	261
$A.D.D._{BC}$	Available-draft drop through boiler and breeching to smoke-conduit connection, in inches water column	516
A_t	Cross-sectional area of a through stay, in square inches	261
$B.t.u.$	British thermal unit	37
c	Distance from axis of moments to extreme fiber	547
C	Constant for stay spacing	159
d	Diameter of chimney flue, in feet	518
d	Internal diameter of shell, in inches	193
D	Diameter of rivet holes, in inches	225
d	Density of gas	609
d_i	Inside diameter of chimney, in inches	555
d_o	Outside diameter of chimney, in inches	555
E	Efficiency	213
$E.D.$	Effective draft, in inches water column	516
f	Factor of safety	196
F	Total force due to wind, in pounds	540
FS	Factor of safety	213
h	Static pressure, inches of water	609
I	Moment of inertia	547
k	A constant	538
L	Internal length of shell, in inches	193
L_d	Length of diagonal stay, in inches	261
L_h	Distance, in inches	260
L_h	Height of chimney, in feet	507
L_{h_c}	Height of center of gravity of projected area of chimney, in feet	542
L_{h_e}	Distance, in feet	551

Symbol	Meaning	Section first used
L_P	Distance from surface supported to center of palm of diagonal stay	261
L_p	Pitch of stays, in inches	262
L_w	Width, in inches	259
L_{wb}	Width of chimney base, in feet	540
L_{wt}	Width of top of chimney, in feet	540
L_U	Width of strip, in inches	236
$\mathbf{N}$	Number of rivets in unit strip	238
P	Pitch of rivets, in inches	225
P	Wind pressure, in pounds per square foot	513
P_2	Atmospheric pressure at altitude of chimney, in pounds per square inch	507
P_{bl}	Longitudinal bursting pressure, in pounds per square inch	208
P_{bt}	Transverse internal bursting pressure, in pounds per square inch	201
p_c	Total maximum pressure, in pounds per square inch	548
p'_c	Pressure, in tons per square foot	545
p'_c	Pressure, in pounds per square inch, due to dead weight of chimney	548
p''_c	Maximum compressive stress, in pounds per square inch	547
P''_c	Maximum stress in chimney wall, in pounds per square inch due to wind	556
P'_c	Compressive stress, in pounds per square inch	555
P'''_c	Stress, in pounds per square inch	558
P_D'	Total static pressure, in inches water column, at base of chimney	507
P''_D	Total static draft which chimney must develop, in inches water column	516
P_{gl}	Internal pressure on shell, in pounds per square inch	204
P_{gt}	Internal pressure, in pounds per square inch, gage	193
P_L	Total longitudinal force, in pounds, on head of shell	204
P_{MAW}	Maximum allowable working pressure, in pounds per square inch	213
P_T	Total pressure, in pounds, tending to rupture shell	193
r	Internal radius of shell, in inches	260
R	Inside radius of shell, in inches	213
S	Transverse stress due to pressure	194
S_c	Unit crushing strength, in pounds per square inch	239
$\mathbf{S}_c$	Total crushing strength of rivet or plate, in pounds	239
$\mathbf{S}_s$	Shearing strength in single shear, in pounds	238
S_s	Unit shearing strength of rivet, in pounds per square inch	238
$\mathbf{S}_T$	Tensile strength of unit strip, in pounds	236
S_T	Unit tensile strength, in pounds per square inch	236

Symbol	Meaning	Section first used
$\mathbf{S}'_T$	Tensile strength of plate between rivet holes, in pounds	237
S_{Tt}	Safe resisting strength against internal transverse bursting pressure, in pounds	196
S_{Tl}	Safe resisting strength against longitudinal pressure, in pounds	206
t	Thickness of shell plate, in inches	194
T_G	Average temperature of chimney gases, in deg. fahr	507
T_0	Temperature of outside air, in deg. fahr	507
TS	Ultimate tensile strength of boiler plate as stamped on shell, in pounds per square inch	213
U_t	Ultimate tensile strength, in pounds per square inch	196
v	Velocity, in miles per hour	538
w	Weight of gas pounds pounds per minute	609
W	Weight steam lb. per hr	311
$\mathbf{W}$	Weight, in pounds	551
$\mathbf{W}_c$	Coal burned per hour, in pounds	518
$\mathbf{W}_t$	Weight of stack and foundation, in tons	545
x	Distance, in feet	551

STEAM BOILERS

DIVISION 1

FUNCTION, CLASSIFICATIONS AND REQUIREMENTS OF THE STEAM BOILER

1. The function of the steam boiler (Fig. 1) is to convert and transfer the chemical energy in the fuel burned, to heat energy in the steam, and thus render it available for use in heating systems and for conversion into mechanical energy by engines or turbines. The fuel may be of fossil origin, such as coal, peat, or petroleum. Or it may be one such as wood,

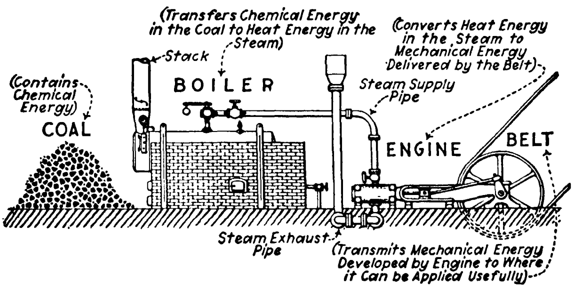


FIG. 1.—Illustrating the function of the steam boiler.

straw, or bagasse, derived from forest or field. (See the author's "Practical Heat.")

NOTE.—As is explained in the author's "PRACTICAL HEAT," the chemical energy in the fuels was imparted to them by the sun.

2. A steam boiler is a closed vessel in which, by the application of heat, water is boiled and thereby converted into steam which is then available for power or heating.

3. The possible classifications of steam boilers are numerous, since the various types and designs may be grouped according to any of a number of different schemes. Thus

they may be classified as to design, construction, method of firing, arrangement of tubes, application, and so on.

4. A classification of boilers into fire tube and water tube is probably the most practical for purposes of general discussion. In this book the boilers of the different types are, in general, considered in accordance with this classification, but no effort has been made to adhere rigidly to it. The classification follows:

- I. Fire tube.
  1. Internally fired.
  2. Locomotive.
  3. Vertical.
  4. Return tubular.
  5. Scotch.
- II. Water tube.
  1. Horizontally inclined straight-tube box header.
  2. Horizontally inclined straight-tube sectional header.
  3. Vertical straight tube.
  4. Vertically inclined straight tube.
  5. Vertical bent tube.
  6. Vertically inclined bent tube.
  7. Porcupine.
  8. Continuous tube.

5. Other steam boiler classifications which are sometimes employed are:

1. Externally fired and internally fired.
2. Horizontal and vertical.
3. Stationary, locomotive, and marine.

6. The requirements of the ideal steam boiler are listed below. It should be understood that the thirteen requirements which follow are for an "ideal" boiler. In practice it may not be possible or feasible to satisfy all of them strictly. In fact the design of a practical boiler must be a compromise. In it all of these desirable requirements should be fulfilled insofar as is attainable at justifiable cost. When conditions merit the installation of a boiler of high cost, then the requirements may obviously be satisfied more nearly than in cases where there can be only a small expenditure for the boiler plant. Where fuel cost or hours of use are low, great refinement in boiler design and the money layout it involves may

not be sound engineering. But in any case, specifications which tend to minimize the risk to human life should be observed. The requirements are:

1. Proper workmanship and simple construction, using materials which experience has shown to be the best, thus avoiding the necessity of early repairs.

2. A mud drum or header to receive the impurities deposited from the water and so placed as to be removed from the action of the fire.

3. A steam and water capacity sufficient to prevent any material fluctuation in steam pressure or water level.

4. A water surface, for the disengagement of the steam from the water, of sufficient extent to prevent priming.

5. A constant and thorough circulation of water throughout the boiler so as to maintain all parts at as nearly the same temperature as possible.

6. A form of construction which will obviate, insofar as is humanly possible, the liability of disastrous explosions.

7. A great excess of strength over any legitimate strain. The boiler should be so constructed as to be free from strains due to unequal expansion. If possible, joints exposed to the direct action of the fire should be avoided.

8. A combustion chamber so arranged that the combustion of gases started in the furnace may be completed before the gases escape to the boiler.

9. A disposition of the heating surface, relative to the direction of flow of the gases of combustion, that will insure the greatest possible transfer of heat to the water in the boiler.

10. Ready accessibility to all parts for cleaning and repairs.

11. Proportionment for the work to be done and capability of operating to its full rated capacity with the highest attainable economy.

12. An auxiliary equipment comprising the very best gages, safety valves, and other fixtures.

13. A setting that will insure maximum efficiency of combustion, minimum maintenance and that will obviate to the highest attainable degree losses of heat by radiation and impairment of the furnace efficiency by the infiltration of air.

QUESTIONS ON DIVISION 1

1. State, in general terms, the function of a steam boiler.

2. How is the energy which is made available by a boiler principally utilized?

3. Give an example of a fossil fuel.

4. Give an example of a vegetable fuel.

5. Whence comes the energy that resides in fuels?

6. Give the conventional definition of a steam boiler.

7. What principal attributes of steam boilers determine their classification?

8. Into what two general divisions may steam boilers be conveniently separated?

9. To which of these divisions does the locomotive form of stationary boiler belong? The dry-back Scotch boiler? The return-tubular boiler?

10. What are the principal requisites to be fulfilled in the building and installation of steam boilers?

11. What considerations determine, in specific cases, the degree in which requisites of the "ideal" boiler shall be approximated?

12. What is the paramount consideration in all cases?

DIVISION 2

EVOLUTION OF THE STEAM BOILER

7. Steam Boilers Are of Ancient Origin.—They have been used for various services and in many forms since remote times. But prior to the eighteenth century none of the devices developed had, as measured by present-day standards, any practical value. However, it is desirable to consider for a moment some of the boilers of the pioneer types so that the reader may be conversant with the evolution which has occurred.

8. The first boilers were of Greek and Roman origin, so it is believed. They were employed several hundred years prior to the Christian Era. They (Figs. 2 and 3) were small affairs, and were employed for warming water, for heating, and for household services. The boiler of Fig. 2, recovered from the ruins of Pompeii, constitutes a typical example. It was of cast bronze and was evidently of the "internally fired" type. The grate (Fig. 3) comprises sheet-bronze tubes opening and brazed into the water leg of the boiler. From this it is evident that the water-tube principle is an old one.

NOTE.—When in use, the boiler proper was supported on an ornamental tripod (Fig. 2) which permitted the air required for combustion to enter from beneath. Three gas vents were arranged from the furnace chamber through the water leg to provide an exit for the products of combustion. The actual Pompeiian boiler was embellished with finely wrought ornaments and miniature sculptures.

9. The Boiler Used with Hero's Engine Was the First Recorded as Doing Mechanical Work.—It was made about 130 B.C. It consisted merely of a hemispherical caldron of ornamental design which was heated by a lamp arranged under it.

NOTE.—Hero's engine was a hollow sphere, mounted and revolving on trunnions, from which extended two nozzles with right-angled turns

at their ends. Steam from the boiler was admitted to the interior of the sphere through one of the trunnions. This steam, in issuing from the nozzles, caused the sphere to rotate by virtue of the same principle as that on which the reaction turbine operates.

10. Florence Raivault's steam bombshell (Fig. 4) was used by that experimenter in 1605 to determine the disruptive force of steam. The spherical shell consisted of hollow copper castings. They were made with walls of varying thickness. Each shell had a single orifice. The experiment consisted in filling a shell with water, plugging the orifice, and applying heat.

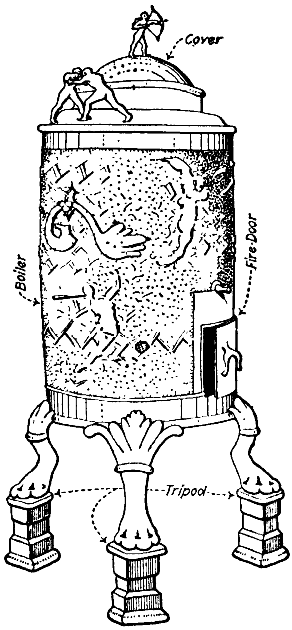


FIG. 2.—Pompeiian domestic water heating boiler (A.D. 79).

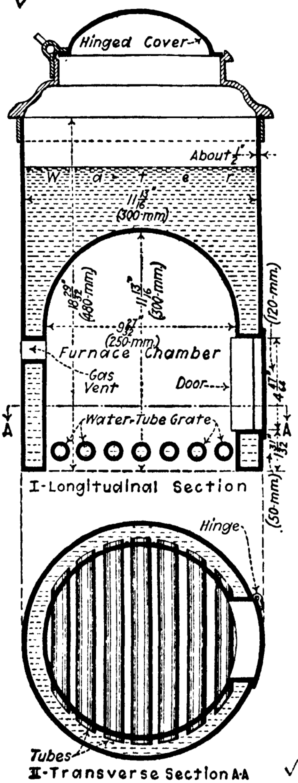


FIG. 3.—Details of the bronze Pompeiian boiler. (Legs not shown.)

11. Giovanni Branca's boiler was made about the year 1629. Branca was an Italian physicist. His invention consisted of a casting in the shape of a hollow human head and trunk. In the mouth of the figure was a nozzle. The hollow casting was filled with water and a fire was built around it. The steam, which then issued from the nozzle, impinged on a wheel having vanes around its periphery. Thus the wheel was caused to turn, and by means of gears its power was transmitted and caused to do work.

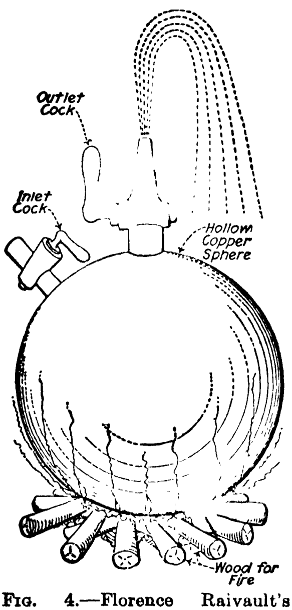


FIG. 4.—Florence Raivault's steam bombshell (year 1605).

12. Denis Papin's boiler (Fig. 5) was used in 1680 and is recorded as a "digester" or apparatus for converting the bones of cattle into a gelatinlike substance. The process required steam at a very high temperature. To obtain the requisite degree of heat, Papin developed a steam pressure of 1,500 lb. per sq. in. He appears to have been the first to produce steam at a pressure greater than 100 lb. per sq. in. This boiler was fitted with a lever safety valve. It is asserted by some that Papin was the inventor of the safety valve.

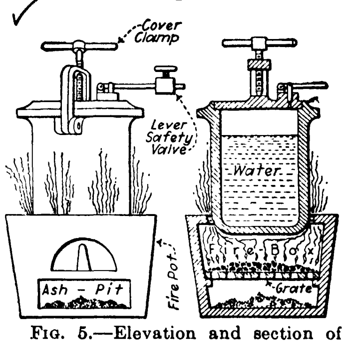


FIG. 5.—Elevation and section of Papin's boiler and safety valve (1680).

13. Savary's boiler (Fig. 6), so named after its constructor, was made in 1698. This apparatus utilized the phenomena

of both condensation and pressure for elevating water. The shell was spherical in shape. This, apparently, is the pioneer example of the application of a specially designed setting and furnace for a steam boiler.

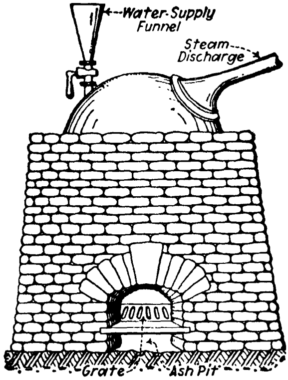


FIG. 6.—Savary's spherical boiler (year 1698).

14. Savary's improved boiler (Fig. 7), constructed in 1702, was an evolution from his original invention. In this the boiler proper was, so it is believed, a hollow cylinder having dished ends.

15. Dr. Desagalier's spherical boiler (Fig. 8) was built in 1718. In this design the hot gases from the fire were caused to circulate in a spiral flue around a spherical water vessel. This marked an advance in furnace construction inasmuch as there was not, as far as is known, any attempt made in the design

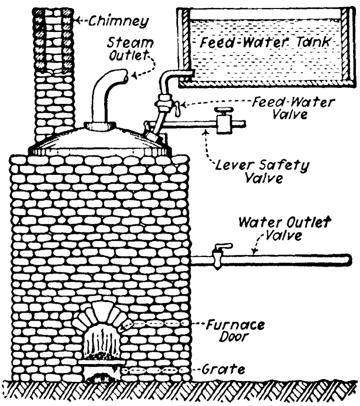


FIG. 7.—Savary's improved boiler (year 1702).

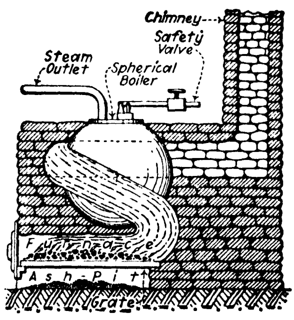


FIG. 8.—Desagalier's spherical boiler (year 1718).

of the earlier boilers to guide the hot gases through paths where they would be most effective in evaporating the water.

16. Newcomen's Mushroom-shaped Boiler Was the First Real Steam Generator (Figs. 9, 10, and 11).—It was built in 1705 and was used to generate steam for Newcomen's atmospheric engine. Previous inventors had inclined generally to the spherical form of vessel as affording maximum strength. But as the engine developed by Newcomen was operated with steam at a very low pressure, he was able to depart somewhat

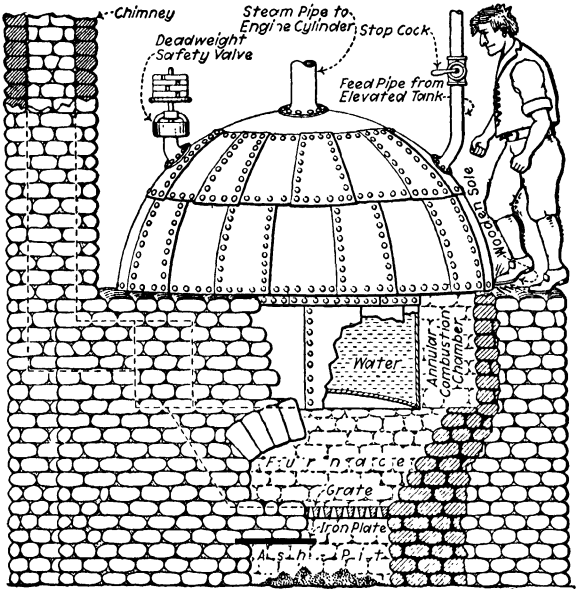


FIG. 9.—Newcomen's mushroom-shaped boiler.

from the spherical form. Hence he designed a boiler that would give a greater proportional area of heating surface and so insure better economy.

17. The constructional details of Newcomen's boiler, as illustrated in the contemporaneous prints, are reproduced in Figs. 9, 10, and 11. The upper half was hemispherical. The lower half was cylindrical with a concave furnace, or crown, sheet. The spherical dome overhung the cylindrical shell beneath. It thus provided a means of suspension in the

setting which permitted the arrangement of an annular combustion chamber around the shell. This boiler was, presumably, braced with some sort of diagonal or gusset stays which secured the flat ring or base sheet to the dome.

18. The accessories of these primitive boilers included usually a dead-weight safety valve. But they were often without gages for indicating the water level and steam pressure. The height of the water in the boiler was determined by the sound, hollow or otherwise, as given out by the boiler sheets when struck with a club or kicked with the wooden-soled brogans worn by the attendant. The simmering of the safety valve indicated when the fire was sufficiently hot. The steam outlet was in the crown of the dome. The water was fed in from an elevated tank by gravity. A round manhole, which had its cover fastened on the outside with stud bolts, was provided in the dome.

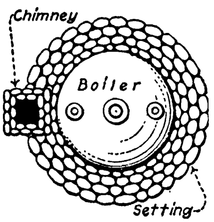


FIG. 10.—Top view of Newcomen's mushroom-shaped boiler.

19. The Haystack Boiler Came into Use about the Middle of the Eighteenth Century (Figs. 12 and 13).—This was probably introduced by Smeaton and was a modification of the mushroom boiler. It was designed to provide greater strength. This was effected by eliminating the flat ring sheet, which formed the base of the dome, and by approaching throughout more nearly the spherical form. Its setting was similar to that of the mushroom boiler. An exception was that the furnace wall was drawn in near the bottom to permit the flange of the crown sheet to sustain the entire weight. Access from the furnace to the annular combustion chamber was provided by three or four openings (*O*, Fig. 12) in the masonry.

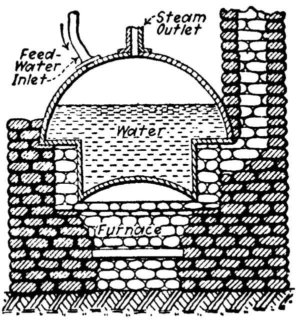


FIG. 11.—Section of Newcomen's boiler and setting (year 1705).

NOTE.—These "haystack" boilers, so it is believed, were sometimes braced by tying the dome sheet to the furnace sheet with through tension-rod stays. But they were generally constructed without stays. Many boilers of this type continued in use in England during the first quarter of the last century.

20. The equipment of the haystack boiler (Fig. 12) shows improvements over that of its predecessor (Fig. 9). The

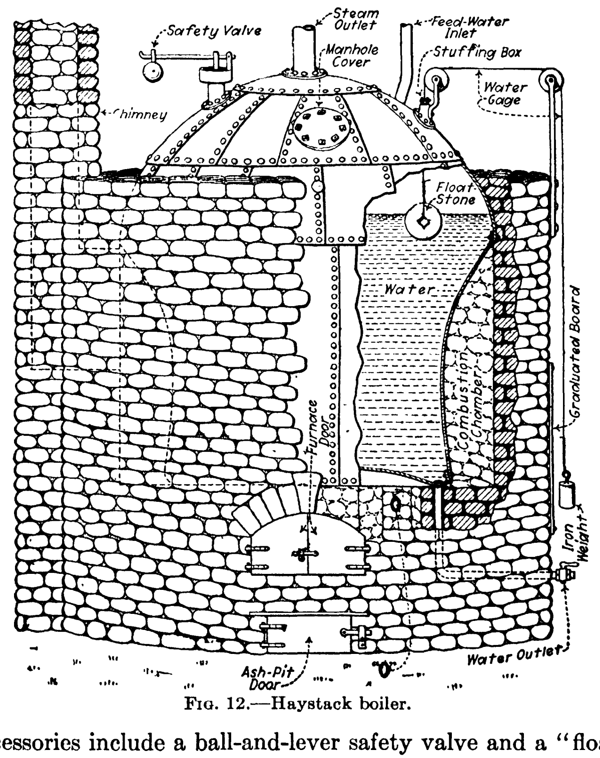


FIG. 12.—Haystack boiler.

accessories include a ball-and-lever safety valve and a "floatstone" for showing the water level. A floatstone consisted usually of a worn-out grindstone hanging on a copper wire on the inside of the boiler. The wire passed out through a stuffing box and ran over two pulleys to an iron weight. The stone rose and fell with the water level. The stage of water was indicated by the position of the weight in relation to

graduations on a board in front of which the weight traveled up and down. A water-outlet pipe with a stopcock was also provided.

21. Watt's Wagon Boiler Was the First Radical Departure from the Spherical Form.—This boiler (Fig. 14) was designed by James Watt in about 1765. The upper half of this boiler was semicylindrical. The lower half was box shaped with concave sides. The bottom was concave and the ends were flat. The masonry was closed in at the upper and lower edges of the concave side sheets so as to form flues along the sides. The two side flues were joined at the front end by a duct which was hollowed out in the masonry. They were separated at the back by a baffle wall that divided the chimney from the back connection. The products of combustion thus swept along the bottom sheet to the back end (about as

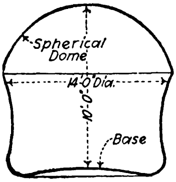


FIG. 13.—Diagrammatic sectional elevation of haystack boiler.

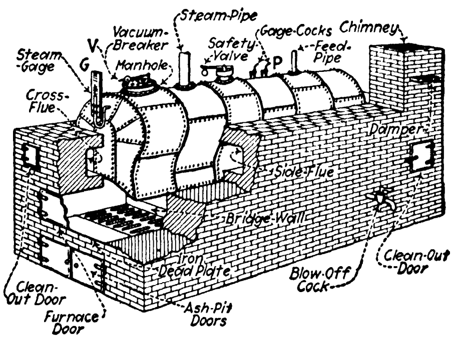


FIG. 14.—Watt's wagon-top boiler.

shown in Fig. 15 for the egg-end boiler), returned through a side flue to the front end, crossed to the opposite side flue, and passed thence to the chimney.

NOTE.—It is probable that these "wagon" boilers were sometimes braced by tying the side sheets together with stay rods. But they were generally without braces of any kind. None of them could handle safely a pressure in excess of about 10 lb. per sq. in.

22. The accessories of Watt's boilers included some appliances which were absent in the designs devised by his predecessors or which were used by them in cruder forms. Notable among Watt's fittings are the steam gage, the water-gage cocks and pipes and the vacuum breaker. The steam gage *G*

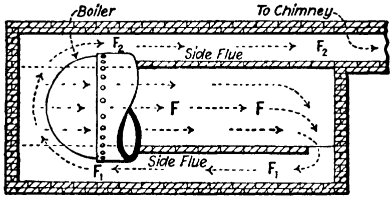


FIG. 15.—Wheel-draft circuit of the egg-end boiler.

was merely a brass U-tube screwed into the front end of the boiler shell. It contained a quantity of mercury. A wooden float, which was carried on the surface of and rose and fell with the mercury column in the outer leg of the U-tube, supported a slender arrow. Thereby, the pressure was indicated on a graduated scale which was attached to the tube.

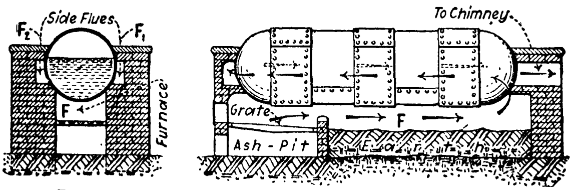


FIG. 16.—The egg-end or cylinder boiler (year 1790).

The water-gage pipes *P* extended down into the boiler. One of them terminated in the steam space an inch or two above the proper water level. The lower end of the other extended into the water space to a depth of about 2 in. below the normal water level. The vacuum breaker *V* was a valve which opened inwards. It was designed to prevent collapse of the structure by the external pressure of the atmosphere when after being in service the boiler was permitted to cool down.

23. The Egg-end or Cylindrical Boiler (Fig. 16) Was the Next Development.—With the extended use of steam power during the closing years of the eighteenth century, there came

a corresponding increase in the pressure required for the newer applications. Coincidently there developed a demand for stronger steam-generating apparatus. About this time the oddly shaped shell of the wagon-top boiler lost its vogue and was superseded by a shell of cylindrical form. Further-

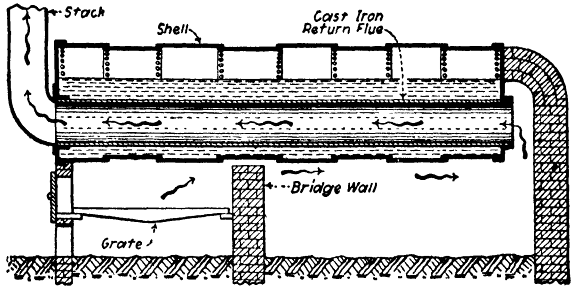


FIG. 17.—Longitudinal section of Evan's return-flue boiler (year 1800).

more, the flat wrought-iron headplates of the wagon boiler were displaced by thick hemispherical cast-iron heads, which tended to provide a boiler of somewhat oval or egg form (Fig. 15). However, in its settings and accessories, the egg-end boiler retained all of the essential characteristics of its immediate predecessor, the wagon-top. These inherited fea-

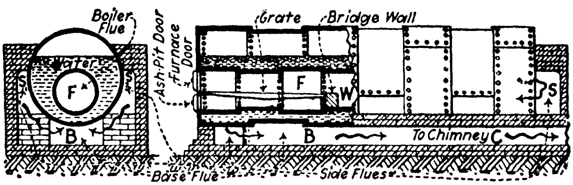


FIG. 18.—Transverse and longitudinal sections of the Cornish boiler (year 1802).

tures included the arrangement of the side and cross flue (Fig. 15) for creating a "wheel draft," as the circuit of the furnace gases around the boiler was termed.

24. The idea of putting the return flue inside of the shell of the boiler instead of running it around outside was first conceived by an American engineer, Oliver Evans, so it is believed. His invention (Fig. 17) was the direct forerunner of all sub-

sequent designs of horizontal return-tubular and flue boilers. Some of these boilers made in about the year 1800 had cast-iron shells and wrought-iron flues. The diameter of the flue was usually about half the shell diameter.

25. The Cornish boiler (Fig. 18) designed by Richard Trevithick, an English engineer connected with the Cornish mining industry, was the result of a plan to improve boiler economy. Trevithick proposed to locate the furnace inside of the flue. The original Cornish boiler, thus invented by Trevithick, may be regarded as the forerunner of all internally fired boilers of subsequent types. The boiler was built with a single flue sufficiently large to hold a furnace having a grate area proportioned properly to the available heating surface. The furnace was made by mounting the grate bars on bearing bars, which were secured crosswise to the flue, at the front and rear ends of the furnace. A bridge wall W to hold the fire on the grate and to form an ashpit was erected at the rear of the furnace.

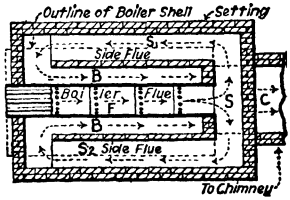


FIG. 19.—The Cornish boiler split-draft circuit.

NOTE.—The current of the furnace gases (Fig. 19) passing out of the flue at the rear end, S, divided and returned through side flues, S_1 and S_2, to the front end. From thence they flowed through a single flue B along the bottom of the shell and into the smoke conduit C at the rear.

26. The first water-tube boiler which embodied the fundamental structural details of modern boilers of this class was the invention of James Barlow about the year 1793. This boiler (Fig. 20) had front and rear water legs. It consisted of a cubical wrought-iron box having within it a smaller boxlike structure. A number of straight tubes extended across from side to side within the smaller box. The preponderance of unstayed flat surface, the exposed top sheet of the fire box, and the restricted water circulation were all obvious disadvantages of this design. It was, perhaps, safe for the pressure commonly employed in its time—3 to 7 lb. per sq. in.

27. A Boiler of the Porcupine Type Was Devised by James Cox Stevens (Fig. 21) in 1804.—In it a number of blind or

pocket tubes were secured in the inclined front and rear faces of a cast-iron water chamber W having a rectangular cross section. The products of combustion in passing to the

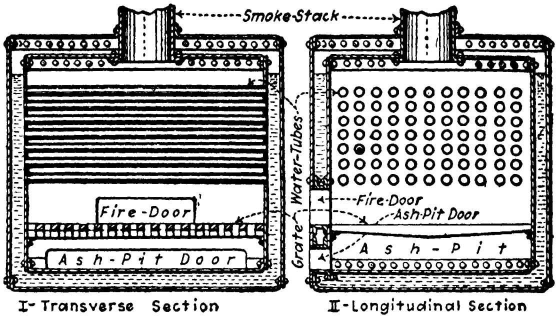


FIG. 20.—Longitudinal and transverse sections of Barlow's horizontal water-tube boiler (year 1793).

chimney S circulated among the tubes B. The water chamber served as a baffle wall, to deflect downward the current of gases through the front bank of tubes and up through the rear bank. A tall steeple-shaped steam dome D, made of wrought-iron

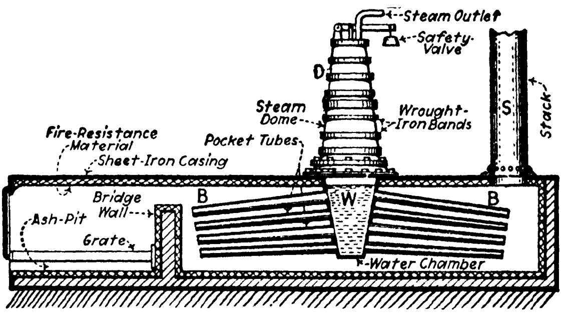


FIG. 21.—Illustration of Steven's porcupine boiler (year 1804).

plates and strengthened with wrought-iron bands shrunk on outside, was bolted to the top of the water chamber.

NOTE.—The Stevens porcupine boiler was built for a working pressure of 50 lb. per sq. in. It marked a definite advance in the design of high-pressure steam-generating apparatus. Its most serious defect appears

to have been the impossibility of adequate water circulation in the tubes. This resulted in the tube ends filling with sediment and burning off.

28. Wilcox's water-tube boiler (Fig. 22) was invented in 1856. This boiler incorporated the general feature of Barlow's (Fig. 20) early attempt. But it was vastly superior both in design and operation. The water leg extended entirely around the firebox. A nest of water tubes, which had a downward inclination, traversed the firebox from front to rear. The crown sheet of the firebox was braced with sling stays which were secured to the semicylindrical top sheet. The inner and outer sheets of the water leg were tied together with stay bolts. From this elementary design were evolved the

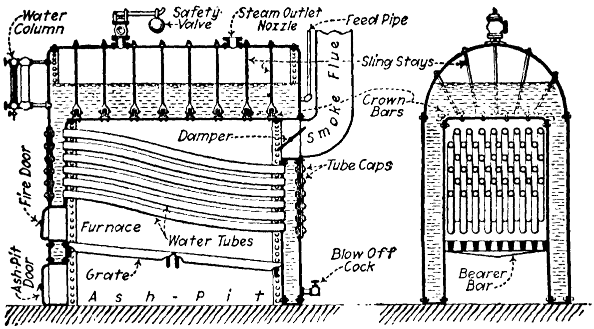


FIG. 22.—Wilcox's boiler with inclined water tubes (year 1856).

various classes of modern water-tube boilers which have inclined tubes that connect front and rear headers or legs.

29. The first sectional boiler having inclined water tubes was made by George Twibill (Fig. 23) in 1865. A nest of inclined wrought-iron tubes traversed the combustion chamber from side to side. The separate fore-and-aft rows of tubes connected, at each side, to horizontal manifolds or headers. These headers were attached, front and rear, to standpipes which carried the steam to two superimposed drums above. One drum was inside and the other outside the masonry. The entrained water was collected in the lower drum and passed thence to the lower set of manifolds.

30. The Babcock and Wilcox sectional water-tube boiler of early form, as shown in Fig. 24, was built about 1870. In it

were represented the essential constructional principles which were developed and refined in the later designs. The tubes were of wrought iron. The vertical header sections were

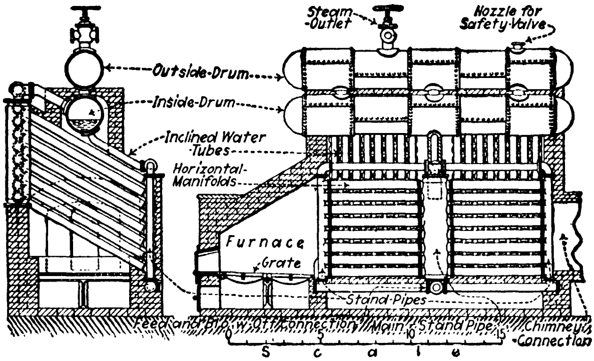


FIG. 23.—Twibill's sectional water-tube boiler (year 1865).

of cast iron. The two were joined together in the foundry by laying the tubes in the mold and casting on the headers. A vertical baffle erected midway caused the gases of com-

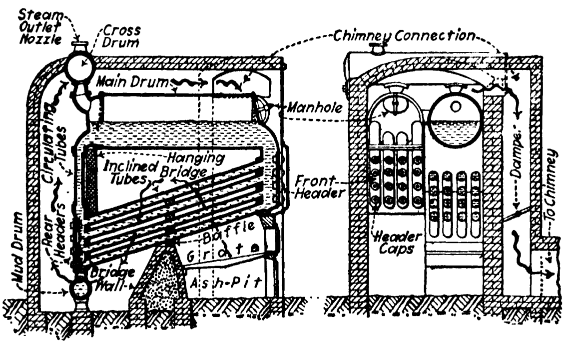


FIG. 24.—Early form of Babcock and Wilcox boiler (year 1870).

bustion to make two passes through the nest of tubes. The external wall of the setting was built to form a connection to the chimney.

QUESTIONS ON DIVISION 2

1. At what period did boilers of practical use, as viewed from a modern standpoint begin to appear?

2. (*a*) What nations of antiquity built the first boilers? (*b*) What were the uses of boilers in ancient times? (*c*) In what antique construction was the water-tube principle first revealed?

3. Describe the first recorded use of steam for developing mechanical work.

4. Was the "steam bombshell" a boiler or an infernal machine?

5. Describe the first recorded use of steam for producing mechanical motion by reaction.

6. To whom is ascribed the invention of the lever safety valve?

7. To whom is ascribed the first definite adaptation of the furnace and setting to the boiler structure?

8. What was the form of Savary's 1702 boiler?

9. Whose was the first recorded attempt at realizing the full steam-making effectiveness of a boiler?

10. (*a*) To what structural shape did primitive steam boilers generally conform? (*b*) Why were they so shaped? (*c*) What was Newcomen's purpose in departing from conventional practice in this regard? (*d*) What circumstance favored him in the adoption of a less stable shape?

11. Describe the structural details of Newcomen's boiler.

12. (*a*) What was a common expedient for finding the water level in the mushroom-topped boiler? (*b*) How was the water replenished? (*c*) How did the attendant know when the fire was hot enough?

13. (*a*) Describe the structural details of the haystack boiler. (*b*) In what respect was the haystack form of construction an improvement on the preceding type?

14. Describe a floatstone water gage.

15. (*a*) Describe the wagon boiler. (*b*) What advantage, over the preceding type, was the wagon boiler designed to secure? (*c*) What was the maximum safe pressure for a wagon boiler?

16. (*a*) Name the principal appliances used on Watt's boilers. (*b*) Describe the steam gage. (*c*) The vacuum breaker. (*d*) The water gages.

17. (*a*) At what period did the egg-end boiler come into use? (*b*) What were the circumstances of its adoption as the prevailing type? (*c*) In what respect was the egg-end boiler an improvement on the wagon boiler? (*d*) Explain the meaning of "wheel draft"?

18. (*a*) To whom is ascribed the invention of the return-flue boiler? (*b*) What materials were used in the early boilers of this type? (*c*) What was the usual ratio of shell diameter to flue diameter?

19. (*a*) Where and by whom was the Cornish boiler first used? (*b*) What incentive led to the design of this boiler? (*c*) How were the gas passes arranged?

20. (*a*) Describe Barlow's water-tube boiler. (*b*) What were the outstanding disadvantages of this boiler?

21. (*a*) Describe Steven's porcupine boiler. (*b*) What was the principal objection to this design?

22. What outstanding feature of Wilcox's 1856 boiler distinguishes it as the prototype of modern horizontal water-tube boilers?

23. Describe the general arrangement of the principal elements of the first sectional boiler with inclined water tubes.

24. How were the tubes joined with the headers in the early forms of Babcock & Wilcox boilers?

DIVISION 3

STEAM GENERATION AND BOILER CAPACITY

31. Heat Must Be Transmitted from the Fire in the Furnace to the Water in the Boiler When Steam Is to Be Generated.—Three methods (Fig. 25) of heat transmission are utilized in getting the heat to the water. These are (1) radiation, (2) convection, (3) conduction.

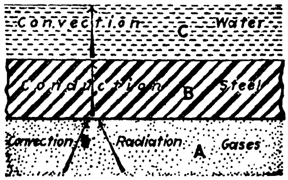


FIG. 25.—Transfer of heat from fire to water.

NOTE.—The three methods of heat transmission enumerated in the preceding section do not, necessarily, act in independent succession. It has been found that all three may operate in unison in the three stages (*A*, *B*, and *C*, Fig. 25), of transmission. Methods of heat transmission are discussed in the author's "Practical Heat."

32. The Resistance to Heat Transfer from the Fire to the Water Should Be a Minimum.—Experiment has shown that a large part of such resistance is due to stagnant or quiescent gas which clings to the heating surfaces. A body of gas in this condition is a good insulator. Provision should be made for producing a rapid general movement of the gases through the combustion chamber and gas passages. Movement may thus be imparted to bodies of gas which might otherwise remain stagnant. Hence their transference of heat by convection will be improved.

NOTE.—Stagnation of furnace gases may be avoided by forcing the gases through restricted passages. This may be accomplished by means of baffles which limit the areas of the passages. The gases are thus constrained to travel a greater distance and at a higher velocity than they would otherwise. In locomotive and some stationary boilers the fire tubes are made as small as practicability will permit. Forced or induced draft is then applied to cause the hot gases to travel at high velocity through the restricted passages thus afforded.

33. Stagnant Water Adjacent to the Heating Surface Prevents Free Flow of Heat to the Mass of Water in the Boiler.—To insure adequate heating of the water and to prevent overheating of the boiler metal, water circulation (Fig. 26) must be provided. This is secured by a structural arrangement which permits the more dense portion of the water to descend through one part of the boiler. There it displaces that portion made less dense by reason of the steam bubbles generated. The hotter portion is thus constrained to flow

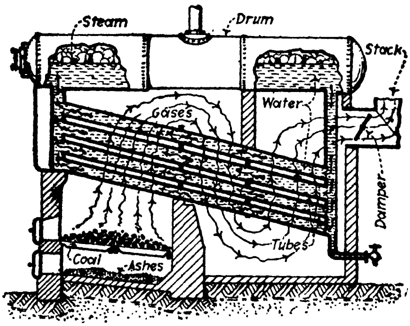


Fig. 26.—Showing water and gas circulation in Babcock and Wilcox boiler.

upward through another part of the boiler. The more rapid the circulation, the better the heat transfer.

34. Steam Bubbles Resist Free Flow of Heat.—Scouring the bubbles from the interior surfaces of a boiler conduces to economy. A rapid water circulation accomplishes this. The rapidity of circulation depends upon the height and relative densities of the downward-flowing column of cool water and the upward-flowing column of mingled hot water and steam.

Note.—The principles of water circulation are illustrated in the author's "Practical Heat."

35. The Steam Space in a Steam Boiler Is the Space above the Water Level.—The steam space should be of sufficient volume to permit complete disengagement of the steam from the water. When the steam flows intermittently from the boiler, the steam space should be larger than when the flow is steady.

36. Capacity of Steam Boilers.—Modern practice is to state the maximum quantity of steam in pounds per hour that the boiler is designed to generate together with the square feet of heating surface. This latter figure is given so that the engineer or purchaser will know how hard the boiler is driven at maximum output, and what decrease in efficiency may be expected at this output.

37. Boiler horsepower used to be used to designate the capacity of all boilers, but in modern boilers it has lost most of its significance. It is not applied to large boilers but is still used to rate smaller units. This unit has no definite mathematical relation to the horsepower unit which is used for expressing the rate of doing work. It is merely an expression for the rate of evaporation of water in a boiler. It is defined as the evaporation of 34.5 lb. of water per hour at a temperature of 212°F. into steam at the same temperature. This is equivalent to an expenditure of 33,472 B.t.u. per hr. Boilers are sometimes rated in boiler horsepower by arbitrarily dividing the heating surface by 10. On this basis modern large units regularly operate at 200 to 400 per cent of rating and have been known to produce during peak loads over 1,000 per cent of their rated output. It is because of this that this method of rating is now being discarded.

38. In computing the heating surfaces of boilers, only the surface exposed to the fire on one side and water on the other should be considered. The A.S.M.E. advocates using the inside diameters of fire tubes and the outside diameters of water tubes as bases of computation for boiler-tube heating surfaces. Heating surface in water walls and in superheaters is usually given separately from the heating surface in the main part of the boiler.

39. Priming and Foaming in a Boiler Are Undesirable (See also note under Sec. 627).—*Priming* is entrainment of water from the boiler with the outgoing steam. *Foaming* is the formation of steam-containing bubbles on the surface of the water.

Note.—Priming may be due to insufficient steam space. The water surface should afford sufficient area for the steam bubbles to ascend from the water without crowding. Forcing a boiler may cause priming.

40. The Quality of Steam Is the Percentage of Water Vapor, as Distinguished from Moisture, Which Is Present in the Total Weight of Steam.—It is often thought of as the *dryness of the steam.* Ordinarily, a boiler delivers steam which contains from 1 to 3 per cent of water. The quality of steam in average practice is, therefore, from about 97 to 99 per cent. The quality of steam may be determined by means of the steam calorimeter, as described in the author's "Practical Heat."

Example.—If half the weight of steam issuing from a pipe is in the form of water particles, the quality of the steam is 50 per cent.

41. Superheated Steam Is Steam That Has a Higher Temperature Than Boiling Water under a Pressure Equal to the Pressure of the Steam.—*Saturated steam* always has the same temperature as the boiling water from which it is generated. If heat is imparted to saturated steam, subsequent to its emergence from contact with the water, it then becomes *superheated steam.* The resulting increase of temperature is called the *degree of superheat.* Steam temperatures as high as 950°F. are in use in central stations and one experimental unit of 10,000 kw. uses steam at 1000°F.

Example.—Saturated steam at a pressure of 100 lb. per sq. in., absolute, has, as shown by the steam tables, a temperature of 327.8°F. If such steam passes through a superheater and emerges therefrom at a temperature of 478.8°F., the superheat is 478.8 − 327.8 = 151.0°F.

DIVISION 4

FIRE-TUBE BOILERS

42. The fire-tube boiler is older than the water-tube boiler. Perhaps for this reason, it is sometimes not looked upon as a modern type. Yet such boilers have a definite place and field of usefulness. When provided with proper combustion equipment, they will produce steam efficiently when operated

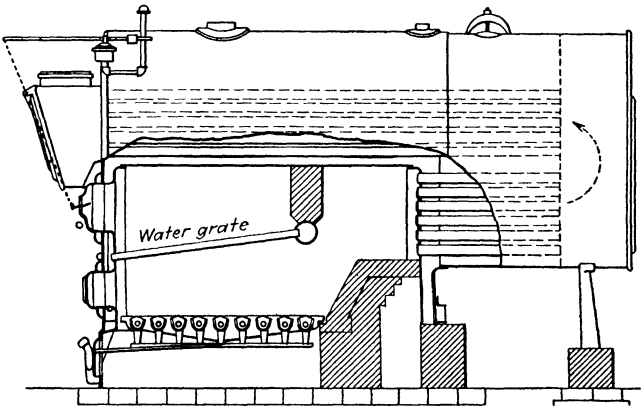


FIG. 27.—Fire-tube boiler with down draft. (*Kewanee Boiler Corporation.*)

within their specified capacity. Their inherent construction limits both size and pressure. Fire-tube stationary boilers are seldom built for more than 150 lb. per sq. in. pressure or for rated outputs over 15,000 lb. of steam per hr. They are seldom operated at more than 150 per cent of rated capacity. These limitations as to size and pressure restrict their use to relatively small plants.

43. Fire-tube Boilers Are Either Externally Fired or Internally Fired (Sec. 4).—Externally fired boilers have a separate furnace built outside of the shell of the boiler. In the inter-

nally fired boilers, the furnace forms an integral part within the boiler structure.

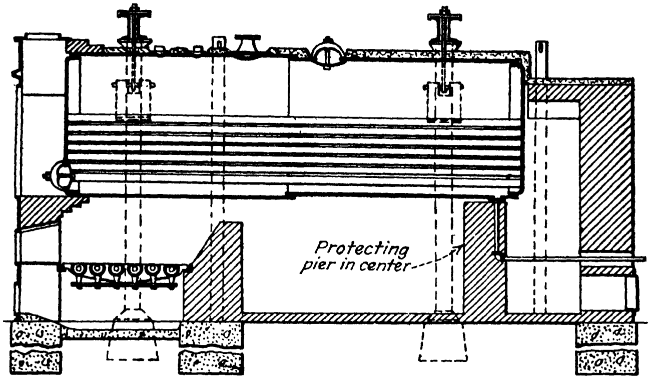


FIG. 28.—Horizontal-return-tubular boiler. (*Kewanee Boiler Corporation.*)

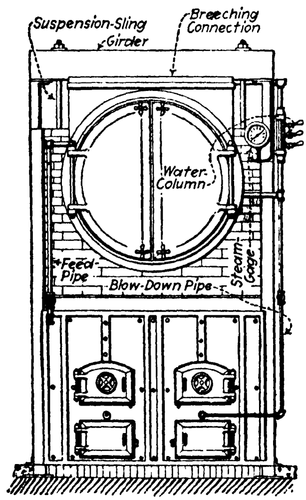


FIG. 29.—Front elevation of horizontal return-tubular boiler.

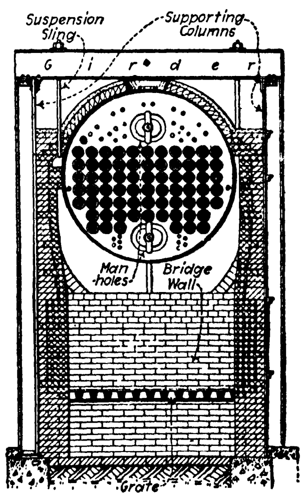


FIG. 30.—Cross-sectional elevation through furnace of horizontal return-tubular boiler.

44. The horizontal return tubular or multitubular boiler (Figs. 27 and 28) is a development of the return-flue boiler.

It was evolved by replacing a few large-diameter flue tubes with many of comparatively small diameter.

Side and front views of a typical boiler of this class are presented in Figs. 27, 28, 29, and 30.

45. Return-tubular boiler, number of tubes, proportions, and thicknesses (see Table I) are from about twenty-four 3-in. tubes, in a 36-in. shell, to eighty-eight 4-in. or one hundred ten 3½-in. tubes, in a 78-in. shell. Shells vary in thickness, as determined by the diameter, from ¼ to ½ in. Thickness of heads varies from ⅜ in., for boilers with 36-in. shells, to $\frac{9}{16}$ in. for those with 78-in. shells. The lengths range from 8 to 20 ft. The steam space in a return-tubular boiler is restricted

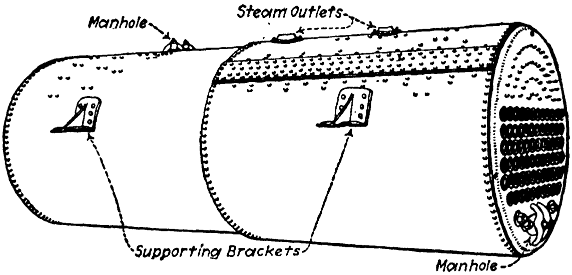


FIG. 31.—Horizontal return-tubular boiler with supporting brackets.

to about one-third of the total volume of the shell. The remaining two-thirds, less the volume occupied by the tubes, is water space. Hence the pressure tends to fluctuate less in this boiler than in the previously described modern types which have relatively less water space.

46. Two Methods of Supporting Return-tubular Boilers Are in Common Use.—One method is (Fig. 30) by suspension from rods secured above to channel beams which bear on steel columns. The rods engage with lugs riveted to the shell (Figs. 28 and 30). The other method (Fig. 31) employs brackets which are riveted to the shell. The brackets rest on iron plates set in the brickwork. With the former method, the expansive action of the boiler shell is unhampered by the restraint which the masonry imposes in the latter case.

47. Advantages claimed for the return-tubular boiler are as follows: (1) It has the greatest evaporative capacity in

TABLE I.—SPECIFICATIONS FOR HORIZONTAL TUBULAR BOILERS*

Longitudinal seams, all butt jointed, triple riveted, 125 lb. working pressure

Horsepower of boiler as rated	80	90	100	110	125	150	165	180	200
Diameter of boiler, in.	60	60	66	66	72	72	72	78	78
Length of tubes, ft.	16	18	16	18	16	18	20	18	20
Number of 3½-in. tubes	54	54	66	66	86	86	86	110	110
Number of 4-in. tubes	44	44	54	54	70	70	70	88	88
Heating surface for 3½-in. tubes, sq. ft.	791.7	890.6	967.6	1,088.6	1,260.8	1,418 4	1,576.0	1,814.0	2,015.9
Heating surface for 4-in. tubes, sq. ft.	737.2	829.4	904 8	1,017.9	1,172.9	1,319 5	1,466.0	1,658.8	1,843.1
Heating surface for shell, sq. ft.	167.6	188 5	184 3	207 4	201 1	226 2	251.3	245.2	272.4
Thickness of shell, in.	13/32	13/32	7/16	7/16	15/32	15/32	15/32	17/32	17/32
Thickness of heads, in.	½	½	½	½	½	½	½	9/16	9/16
Width of grates, in.	60	60	66	66	72	72	72	78	78
Length of grates, in.	54	60	54	60	54	60	66	60	66
Area of grates, sq. ft.	22.5	25.0	24.75	27.5	27.0	30 0	33.0	32.5	35.75
Diameter of stack, in.	28	28	30	30	34	34	34	38	38
Length of stack, ft.	60	60	60	60	60	60	70	60	70
Gauge of stack	14	14	14	14	14	14	14	12	12
Length of guys, ft.	600	600	600	600	600	600	700	600	700
Diameter of guys, in.	5/16	5/16	5/16	5/16	5/16	5/16	5/16	⅜	⅜
Trimmings									
Size of steam opening,† in.	4	5	5	5	6	6	6	6	6
Size of pop safety valves,† in.	Two 2½‡	Two 3	Two 3	Two 3	Two 3	Two 3½	Two 3½	Two 3½	Two 4
Size of water gage glass and gage cocks, in.	¾	¾	¾	¾	¾	¾	¾	¾	¾
Size of water column connections, in.	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼
Size of blowoff, in.	2	2	2	2	2½	2½	2½	2½	2½
Size of feed and check valves, in.	1½	1½	1½	1½	1½	1½	1½	2	2
Size of steam gage dial, in.	8½	8½	8½	8½	8½	8½	8½	8½	8½

Weights

Horsepower of boiler as rated	80	90	100	110	125	150	165	180	200
Bare boiler, for full-front setting, lb.	12,560	13,790	14,970	16,460	18,530	20,380	22,250	25,480	27,900
Bare boiler, for half-front setting, lb.	13,040	14,270	15,560	17,050	19,355	21,205	23,075	26,505	28,925
Full front, lb.	2,400	2,400	2,590	2,590	2,970	2,970	2,970	3,170	3.170
Half front, lb.	1,140	1,140	1,280	1,280	1,510	1,510	1,510	1,700	1,700
Grates, lb.	1,040	1,420	1,090	1,490	1,145	1,560	1,890	1,705	2,065
Other castings, lb.	380	380	395	395	430	430	430	725	725
Trimmings, lb.	225	225	225	225	235	260	260	335	335
Stack and guys, lb.	1,600	1,600	1,750	1,750	2,000	2,000	2,300	2,800	3,300
Weight complete, full-front setting, lb	18,205	19,815	21,020	22,910	25,310	27,600	30,100	34,215	37,495
Weight complete, half-front setting, lb.	17,425	19,035	20,300	22,190	24,675	26,965	29,465	33,770	37,050

Extras and changes

Four wall plates and rollers, lb.	500	500	500	500	500	500	500	500	500
Four buck bars and cross rods, lb.	575	575	630	630	750	750	750	800	800
Two rear diamond washers with rods, lb.	250	260	250	260	250	260	280	260	280
Fire-door arch liners, lb.	475	475	475	475	475	475	475	500	500

* Boilers manufactured by Kewanee Boiler Company, Kewanee, Ill.

† All openings over 3 in. will be forged steel-flanged nozzles.

‡ With 3½-in. tubes this boiler has two 3-in. safety valves.

TABLE I.—SPECIFICATIONS FOR HORIZONTAL TUBULAR BOILERS.*—(*Continued*)
Longitudinal seams, all butt jointed, quadruple riveted, 150 lb. working pressure

Horsepower of boiler as rated	80	90	100	110	125	150	165	180	200
Diameter of boiler, in.	60	60	66	66	72	72	72	78	78
Length of tubes, ft.	16	18	16	18	16	18	20	18	20
Number of 3½-in. tubes	54	54	66	66	86	86	86	110	110
Number of 4-in. tubes	44	44	54	54	70	70	70	88	88
Heating surface for 3½-in. tubes, sq. ft.	791.7	890.6	967.6	1,088.6	1,260 8	1,418.4	1,576.0	1,814.0	2,015.9
Heating surface for 4-in. tubes, sq. ft.	737.2	829.4	904 8	1,017.9	1,172.9	1,319.5	1,466.0	1,658 8	1,843 1
Heating surface for shell, sq. ft.	167.6	188.5	184.3	207.4	201.1	226 2	251.3	245 2	272.4
Thickness of shell, in.	7/16	7/16	½	½	17/32	17/32	17/32	19/32	19/32
Thickness of heads, in.	½	½	½	½	9/16	9/16	9/16	9/16	9/16
Width of grates, in.	60	60	66	66	72	72	72	78	78
Length of grates, in.	54	60	54	60	54	60	66	60	66
Area of grates, sq. ft.	22.5	25.0	24 75	27.5	27.0	30.0	33.0	32.5	35.75
Diameter of stack, in.	28	28	30	30	34	34	34	38	38
Length of stack, ft.	60	60	60	60	60	60	70	60	70
Gauge of stack	14	14	14	14	14	14	14	12	12
Length of guys, ft.	600	600	600	600	600	600	700	600	700
Diameter of guys, in.	5/16	5/16	5/16	5/16	5/16	5/16	5/16	⅜	⅜

Trimmings

Size of steam opening,† in.	4	5	5	5	6	6	6	6	6
Size of pop safety valves,† in.	Two 2½	Two 2½	Two 3	Two 3	Two 3	Two 3	Two 3§	Two 3½	Two 3½§
Size of water gage glass and gage cocks, in.	¾	¾	¾	¾	¾	¾	¾	¾	¾
Size of water column connections, in.	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼	1¼
Size of blowoff, in.	2	2	2	2	2½	2½	2½	2½	2½
Size of feed and check valves, in.	1½	1½	1½	1½	1½	1½	1½	2	2
Size of steam gage dial, in	8½	8½	8½	8½	8½	8½	8½	8½	8½

Weights

Horsepower of boiler as rated	80	90	100	110	125	150	165	180	200
Bare boiler, for full-front setting, lb	13,350	14,700	16,025	17,660	19,930	21,900	23,970	26,810	29,370
Bare boiler, for half-front setting, lb.	13,830	15,180	16,615	18,250	20,755	22,725	24,795	27,835	30,395
Full front, lb.	2,400	2,400	2,590	2,590	2,970	2,970	2,970	3,170	3,170
Half front, lb.	1,140	1,140	1,280	1,280	1,510	1,510	1,510	1,700	1,700
Grates, lb.	1,040	1,420	1,090	1,490	1,145	1,560	1,890	1,705	2,065
Other castings, lb.	380	380	395	395	430	430	430	725	725
Trimmings, lb.	145	145	255	255	275	275	300	310	310
Stack and guys, lb.	1,600	1,600	1,750	1,750	2,000	2,000	2,300	2,800	3,300
Weight complete, full-front setting, lb	18,915	20,645	22,105	24,140	26,750	29,135	31,860	35,520	38,940
Weight complete, half-front setting, lb.	18,135	19,865	21,385	23,420	26,115	28,500	31,225	35,075	38,495
Extras and changes									
Four wall plates and roller, lb.	500	500	500	500	500	500	500	500	500
Four buck bars and cross rods, lb.	575	575	630	630	750	750	750	800	800
Two rear diamond washers with rods, lb.	250	260	250	260	250	260	280	260	280
Fire-door arch liners, lb.	475	475	475	475	475	475	475	500	500

* Boilers manufactured by Kewanee Boiler Company, Kewanee, Ill.

† All openings over 3 in. will be forged steel-flanged nozzles.

§ With 3½-in. tubes safety valves are one size larger.

proportion to its bulk of all fire-tube boilers which present an area of external heating surface. This is true whether the boilers are externally or internally fired. (2) The water content is divided into thin currents circulating in contact with a multiplicity of tubes. Therefore, the tendency is that the heat of the gases will be transmitted simultaneously to all parts of the bulk of the water. Consequently, the boiler steams readily and responds promptly to overloads. (3) For a given extent of heating surface, it is the least expensive of all boilers which have good evaporative efficiency.

48. Positive disadvantages usually charged to the multitubular boiler are as follows: (1) Its record for years past shows that it is more liable to explosion than the other types now in use. (2) The circular seams with their double thickness of plate are exposed to the direct action of the fire. This introduces an element of weakness. This disadvantage is shared in common by all externally fired shell boilers. (3) The gas of combustion tends to short-circuit through the upper rows of tubes. This renders partially ineffective the heating surface presented by the lower rows. (4) Where the furnace is improperly designed, the gas currents are separated, by the multitubular arrangement, into finely divided streams. Hence with a poorly proportioned furnace, the long tongues of flames from bituminous coals of a very gaseous character become extinguished immediately when they enter the constricted tube areas. Consequently, the incompletely burned gas escapes as smoke and the incandescent carbon particles which are mixed with the flaming gas are deposited as soot. However, this disadvantage is one of furnace rather than of boiler design. (5) The water surfaces of the tubes are difficult of access. (6) Crowding of the water space with tubes tends to impede circulation.

49. The service for which the return-tubular boiler is fitted is really a matter of economics. For some services it may be the most economical type, but for others it may not. The first cost per boiler horsepower is very low. These boilers have for this reason been used extensively, even in situations where an analysis of the conditions on an annual operating-cost basis would not justify their installation.

Ordinarily, they are not built for steam pressures greater than 150 lb. per sq. in. or for capacities greater than 200 boiler hp. Hence they are utilized largely in small hand-fired plants. Experience has shown that the metal in the shell immediately above the fire should not exceed a certain thickness. If it is made too thick, the plates deteriorate quickly, owing to overheating and the consequent crystallization. This limitation of plate thickness determines the maximum pressures and capacities for which the boilers should be built.

Return-tubular boilers are not suited for installations where the only available fuel is coal of a very gaseous character, that is, coal which burns with a long streaming flame. They

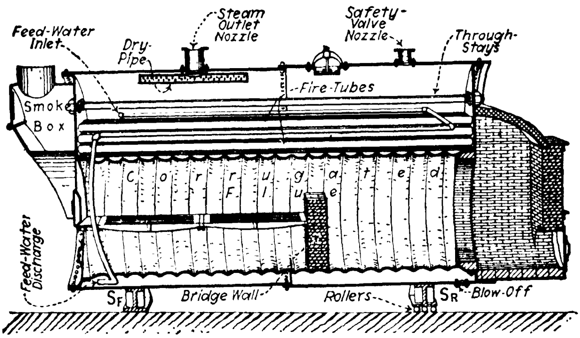


FIG. 32.—Sectional view of internally-fired-multitubular boiler.

give better service with caking and cannel coals. Neither are they adaptable for large chain-grate-stoker installations designed for burning fuels of very low grades.

50. The dry-back variety of the Scotch boiler is an adaptation for stationary plants of the type of boiler popular in marine service in which compactness is a prime consideration. It is a quick-steaming boiler, occupies small space for the power developed, and shows a good economy.

51. Its construction is shown in Fig. 32. The furnace flue is corrugated to provide maximum strength. The shell rests in saddles S_F and S_R (Fig. 32). The rear saddle S_R usually rides on rollers which are so placed as to permit free endwise expansion. By making the back connection external to the boiler [instead of an inseparable part of it, as in the true

Scotch boiler (Fig. 33)], the construction is much simplified. Also the interior of the boiler is thereby more accessible for cleaning. Furthermore, the necessity for bracing much additional flat surface is eliminated. These boilers are now

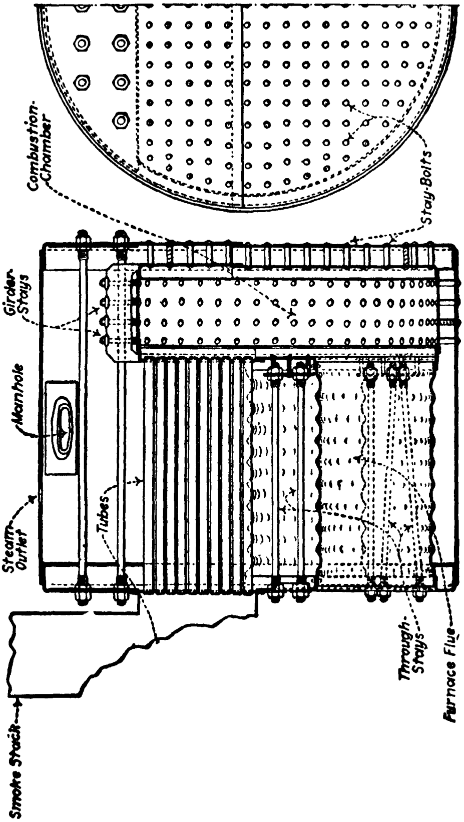


FIG. 33.—Sectional elevation lengthwise of Scotch wet-back boiler.

used to some extent in the United States for temporary installations. They can be transported readily and installed at small expense.

52. The true form of Scotch or drum boiler has the water-back construction, as shown in Figs. 33, 34, and 35. It was developed to satisfy the demand for a very compact absolutely

self-contained boiler. Since these boilers occupy, probably, less volume per unit of power developed than those of any

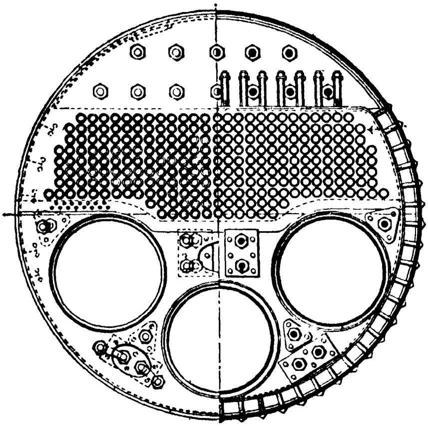


Fig. 34.—Semi-front and cross-sectional elevations of Scotch boiler.

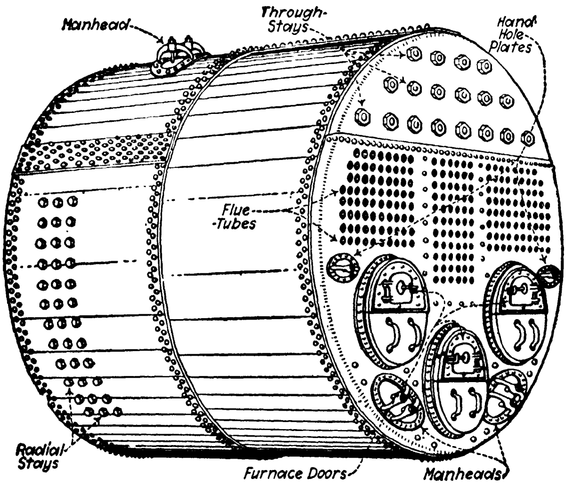


Fig. 35.—External view of typical Scotch boiler.

other type, they are installed in locations where space is very valuable. In the past they have been used widely in marine

service, but more recently have been superseded by water-tube boilers.

53. The construction of the Scotch marine-type boiler is detailed in Figs. 33 to 35 inclusive. They have been built with external shells of diameters up to 20 ft. The usual diameter is from 10 to 15 ft. and the length from 7 to 11 ft. From two to four furnace flues short and of relatively large diameter are used. The grate bars are set just below the horizontal diameter of each flue. Furnace gases pass into a combustion chamber at the rear. Thence they flow through groups of small flue tubes to the chimney uptake at the front end. Safety of this boiler is determined largely by the care that is exercised in staying the extensive areas of flat surface which are presented by the combustion chamber and the external heads. The back sheet of the combustion chamber is secured to the external head with stay bolts. The front sheet is stayed by the furnace flues and tubes. The front- and rear-head areas above the tubes and combustion chamber are also bonded together with through stays. The top sheet of the combustion chamber is braced with girder stays.

54. The leading advantages of the Scotch boiler may be enumerated thus: (1) It occupies the least volume as compared with the power developed of any boiler suitable for general power-plant work. (2) Its free-steaming characteristic renders easy the maintenance of a uniform steam pressure under a varying load.

55. The principal disadvantage of the Scotch boiler is the tendency of the water in it to circulate sluggishly in the region below the furnace flues. This results in the adjacent area of the shell plate remaining relatively cool. Thus strains due to unequal expansion are generated.

56. Internally fired boilers patterned after the fire box type which is used in locomotive design (Fig. 36) are employed largely for portable and semiportable applications. They are used with transient agricultural and sawmill outfits and under similar conditions. Occasionally they are found in stationary plants.

57. The construction of the locomotive-type boiler is shown in Fig. 36. It contains a rectangular firebox having water

legs formed on its sides and ends by the vertical sheets of an outer boxlike construction which encloses the firebox at the sides, front, and rear. The side sheets, top sheet, and rear sheet of the outer box are attached to a cylindrical shell which is filled with flue tubes. Through these tubes the products of combustion flow directly to the stack. The tubes extend from the rear or tube sheet of the firebox to the head of the shell. They are expanded into bored holes at each end. The bottoms of the water legs are closed by riveting the inner and outer sheets to a heavy flanged mud ring. Or, the inner sheet may be flanged outward and

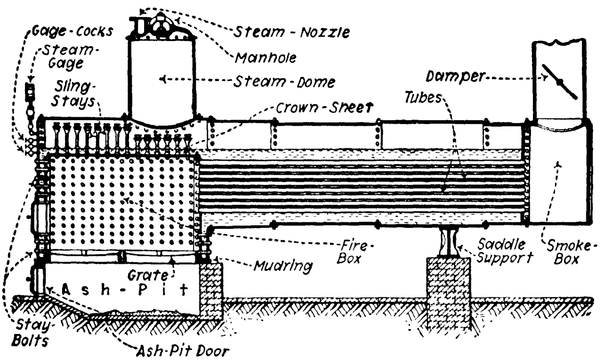


FIG. 36.—Sectional view of locomotive type of boiler as used in stationary practice.

downward to form a riveted joint with the outer sheet. The top of the firebox is flat or slightly arched. It is braced with stays secured to the forward extension of the top half of the shell or "wagon top" as it is sometimes called. The grate bars rest on angles which are fastened with stud bolts to the front and back firebox sheets. Ordinarily, the ashpit or pan is separate from the firebox. But sometimes (Fig. 37), it is formed by providing bottom sheets for the firebox and its external envelope. In this construction, the vertical water leg around the firebox merges with a *water bottom*, as the space between the inner and outer bottom sheets is called. The firebox is entirely surrounded by water, except for the fuel and ashpit openings.

58. The tubes of a locomotive-type boiler for stationary service are fewer in number, but larger in diameter, than those

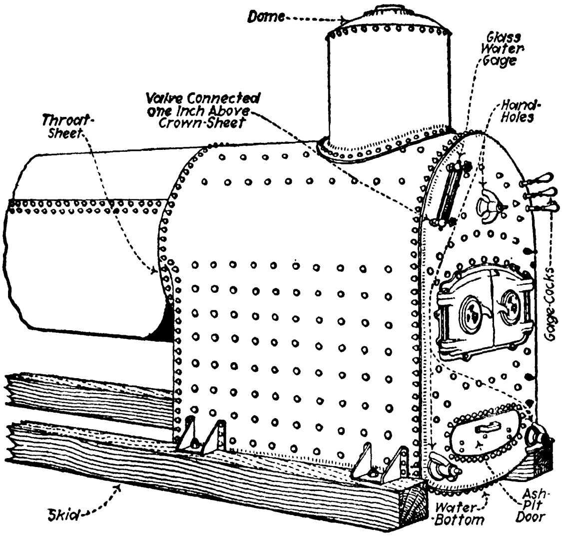


FIG. 37.—Fire-box end of locomotive-style of stationary boiler with water bottom.

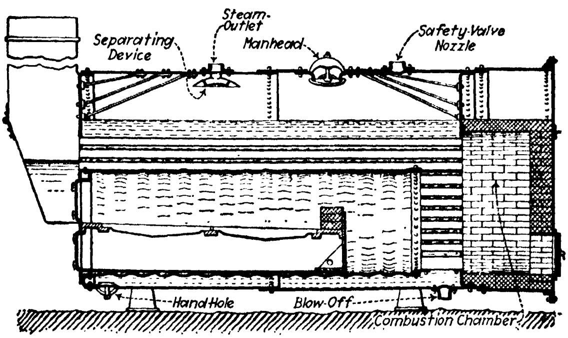


FIG. 38.—Sectional view of "Duplex" boiler.

for a boiler intended for railway service. In a locomotive, quick steaming takes precedence over other desirable qualities. Diameters of 3 and 4 in. are good sizes. The large

tubes reduce the heating surface but conduce to better water circulation.

59. The principal merits claimed for boilers of the locomotive firebox and kindred types are (1) compactness, (2) great steaming capacity, (3) fair economy, (4) mobility.

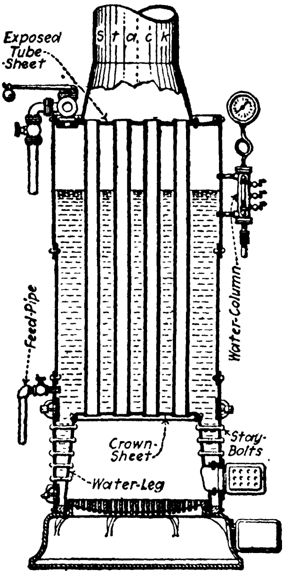


FIG. 39.—Semiportable upright fire-tube boiler having non-submerged tubes.

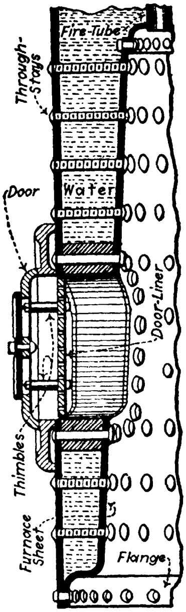


FIG. 40.—Furnace sheet flanged to close water-leg at bottom.

60. The objectionable features most prominent in firebox boilers are (1) the great extent of flat surface which requires bracing, (2) the sluggishness of water circulation which arrangement of their parts induces, (3) the liability of corrosion in the water legs on account of sedimental deposits, (4) the difficulty of reaching the inside for cleaning.

61. A vertical form of firebox boiler which is well adapted for construction outfits in connection with hoisting engines is shown in Fig. 39. It is much used for temporarily located

plants since it occupies but little space, is readily maintained, and easily transported.

62. As to construction, the firebox of a vertical boiler is made with a flat top or crown sheet and usually with a cylindrical ring-sheet roundabout. But sometimes it is built with a ring sheet formed like a truncated cone, as shown in Fig. 39.

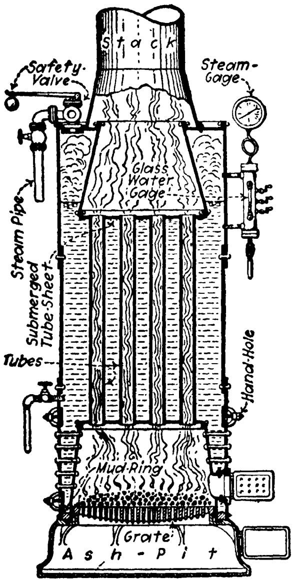


FIG. 41.—Upright boiler with submerged tubes.

A nest of flue tubes extends from the crown sheet to the head sheet of an external cylindrical shell. The tubes are usually 2 in. in diameter. An annular water leg, closed at the bottom with a forged wrought-iron mud ring (Fig. 39) or by flanging the firebox sheet to meet the shell in a riveted joint (Fig. 40), surrounds the firebox. The firebox is stayed to the shell with screwed stay bolts. The tubes serve to stay the crown sheet and the external head. The boiler is set on a cast-iron base which contains the ashpit. Usually the grate bars are supported by an enclosed iron ring fastened to the furnace sheet with studs just above the mud ring. Sometimes, however, they rest on the base (Fig. 39).

63. A Common Defect of Upright Boilers Is Overheating of the Unprotected Tube Ends above the Water Line.—This may result in a loosening of the beaded joints between the tube ends and the tube sheet. To overcome this difficulty, these boilers are often built (Fig. 41) with a submerged tube sheet. The heavier material and construction of the depressed smoke connection which is required with this method are better adapted to resist distortion from overheating than are the relatively light tube ends of Fig. 39.

NOTE.—Great care should be exercised in raising steam in an upright boiler. The fire should be kindled gradually until steam contacts with

the unprotected tube ends above the water line. Never use oily waste or other highly combustible substance in starting a fire in an upright boiler. These will overheat the top sheet and may cause a leak. The submerged-tube type (Fig. 41) is much more reliable in this respect than is the nonsubmerged-tube type (Fig. 39).

64. Objections to the vertical fire-tube boilers are as follows: (1) The steam-liberating surface is small. In the submerged-tube design, the bubbles tend to huddle under the tube sheets. There they form small pockets of steam which produce a geyserlike action in their passage from the water. The result is priming and wet steam. (2) The water circulation is indeterminate and sluggish. (3) Forced firing may cause damage to the crown sheet. This is on account of the extremely rapid evaporation of the water in contact with the sheet. (4) Its operation is apt to involve an element of danger since it contains the least quantity of water in proportion to its steaming capacity of any of the shell boilers. (5) As ordinarily built the short travel of the furnace gas renders it very wasteful of fuel. (6) Access to the water legs for cleaning is difficult. (7) Corrosion of the shell is rapid around the firing door and also below the grates and around the bottom seam at which locations ashes accumulate.

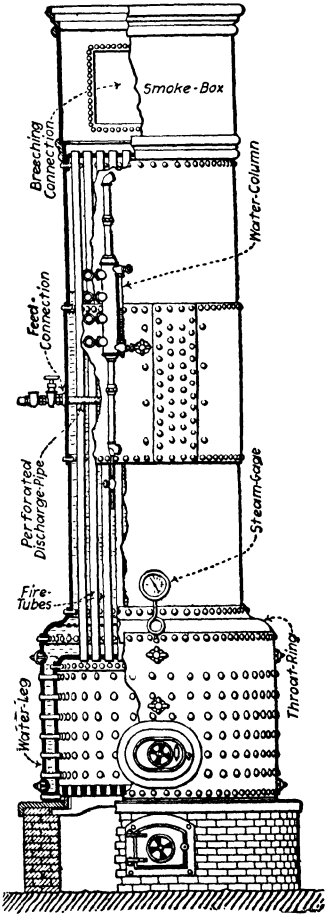


FIG. 42.—Manning vertical fire-tube boiler.

65. The principal merits of the vertical fire-tube boiler are as follows: (1) It is a rapid steamer. Its vertical heating surface gives a decided advantage in this regard. (2) Since it occupies but little space and is transported readily, it is well adapted for temporarily located plants.

66. A Manning vertical fire-tube boiler designed for installation in stationary plants where the available floor space is scant is illustrated in Fig. 42. It is set either on a brick foundation with the ash-pit built in, or on a cast-iron base which rests on a masonry foundation. The fire box is made larger than in boilers of the portable type. This is effected by increasing the diameters of both of the circular sheets which form the water legs. The enlarged section of the external shell is joined to the smaller section above by a double-flanged throat ring. Comparatively small tubes—generally of 2½ in. diameter—are used. They range in length from 12 to 15 ft. The tubes are grouped in four nests (Fig. 43). There is ample cleaning space between. The spaces afford convenient access for washing the top of the crown sheet through handholes in the shell. A perforated extension of the feed pipe inside the boiler delivers the water in finely divided streams among the tubes.

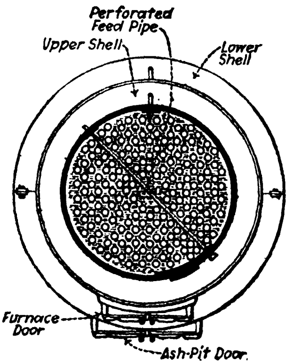


FIG. 43.—Cross-section of tubes in Manning boiler.

67. The advantages claimed for the Manning boiler over ordinary boilers of its class are as follows: (1) A proportionately larger grate area, thus permitting a slower combustion rate per square foot of grate area for a given evaporative effect. (2) Greater tube length gives flexibility under stress of expansion. This permits the upper ends of the tubes to serve as superheating surface without detriment to the tube sheet joints. (3) The offset in the external shell serves to isolate the stresses in the upper and lower sections. (4) Better facilities for cleaning.

QUESTIONS ON DIVISION 4

1. What is a battery of boilers?

2. What is the fundamental difference between internally fired and externally fired boilers?

3. What is a boiler "tube"?

4. If the length and total cross-sectional area of the tubes are the same, which will have the greater area of tube heating surface—a multi-tubular boiler with 3-in. tubes or one with 2.5-in. tubes?

5. What is the average plate thickness in the shells of return-tubular boilers? In the heads?

6. What is the usual ratio of steam space to total shell space in return-tubular boilers? Figured on this basis, how many cubic feet of water would a boiler of 48-in. diameter, 14-ft. length and filled with forty-six 3-in. tubes ordinarily contain?

7. What is the chief objection to bracket support of a shell boiler?

8. Is the general design of the return-tubular boiler conducive to smokeless combustion of bituminous coals? Why?

9. Give a brief description of the Scotch boiler?

10. What is the outstanding advantage of the Scotch boiler? What is its principal defect?

11. What is understood by the terms *dry back* and *wet back?*

12. Describe a vertical fire-tube boiler.

13. What are the merits and disadvantages of the vertical fire-tube boiler?

DIVISION 5

WATER-TUBE BOILERS

68. Inherent design features have permitted a much greater development of boilers of the water-tube type than has been possible with the fire-tube type. Present-day water-tube boilers are built in small and large capacities even up to single units that produce over a million pounds of steam an hour. This type has permitted continued increase in steam pressures

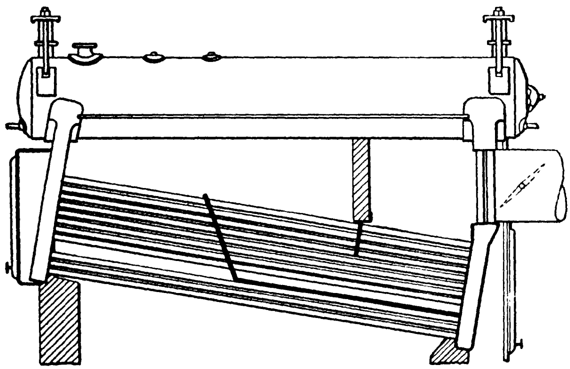


FIG. 44.—Longitudinal-drum water-tube boiler with box headers. (*Murray Iron Works.*)

and many units are now operating at over 1,400 lb. per sq. in. pressure. Superheaters are used with these boilers to raise the temperature of the steam above saturation temperature to final temperatures as high as 950°F. To permit greater evaporation from a given boiler the furnace of the modern unit is surrounded by water-cooled walls that are an integral part of the boiler. In fact general arrangement and design of a modern boiler must go hand in hand with design of the furnace and combustion equipment.

69. Horizontal inclined straight-tube boilers may have either longitudinal or cross drums and may have either box-

headers or sectional headers into which the tubes are rolled. When boxheaders are used, 300 lb. per sq. in. is about the maximum economical steam pressure, and capacity is limited by the connection between the header and drum.

70. Longitudinal-drum straight-tube boilers (Figs. 44 and 45) have inclined tubes expanded front and rear into water

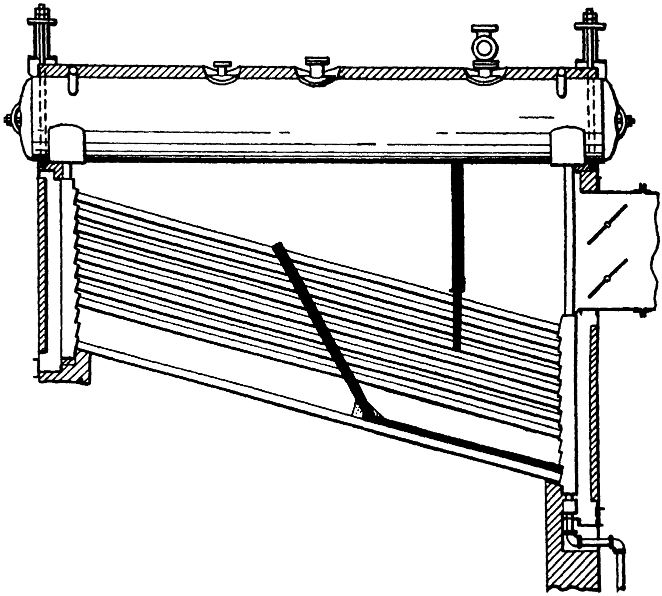


FIG. 45.—Straight-tube longitudinal-drum sectional-header boiler. (*Foster Wheeler Corporation.*)

legs that connect to a horizontal drum parallel to the tubes. Small boilers have one drum. Larger boilers, more than about 14 tubes wide, have two or more parallel drums. In diameter the drums vary from about 30 to 38 in. and in length from 17 to 22 ft. The tubes are arranged in staggered vertical rows so as to make the furnace gas zigzag across them. The complete boiler structure is hung from supporting steel entirely independent of the brickwork.

71. Box-type water legs are formed of steel plate riveted together to form boxes about 8 in. deep. **The water leg at**

the high end of the tubes has a flanged semicircular throat so it may be riveted directly to the drum or drums as in Fig. 46. At the low end the water leg is connected to the drum by short tubes expanded into the top of the boxheader and into a throat connection riveted onto the drum. Both headers are inclined so that the tubes enter the tube sheet at a right angle. Handholes are provided opposite each tube through which the tubes are inserted, expanded, and later cleaned.

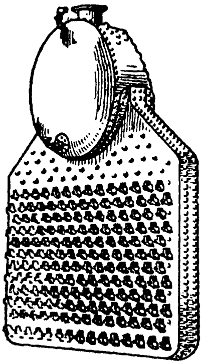


FIG. 46.—Typical box header.

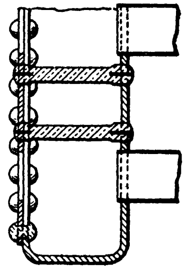


FIG. 47.—Cross section of box-header showing staybolts.

Handhole and tube sheets are tied together with stay bolts as in Fig. 47.

72. Sectional-type water legs (Figs. 48, 49, and 50) are sinuous-shaped wrought-steel headers into which usually one vertical row of staggered tubes are expanded. An exception is shown in Fig. 49 which accommodates two vertical rows of tubes. Likewise in this case one handhole serves four tubes whereas in the other headers shown in Figs. 48 and 50 each tube is served by a single handhole. Sectional headers are vertical, and each is connected to the drum by one or two short tubes. They are seldom more than 24 tubes high though boilers 43 tubes high have been built.

73. Straight-tube boilers usually have 4-in. diameter tubes, but when the header (Fig. 49) is used 3-in. diameter tubes are employed. Some manufacturers reduce the size

of tube as pressure increases using $3\frac{1}{2}$ in. above 500 lb. per sq. in. and $3\frac{1}{4}$ in. for pressures above 900 lb. per sq. in. Heat transfer is somewhat better with small tubes, but as tube diameter decreases, the number of headers required in a boiler of given surface increases. Headers are costly, hence

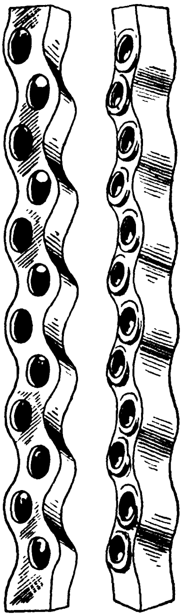


Fig. 48.—Front and back view of a vertical header with elliptical handholes.

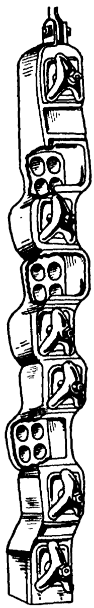


Fig. 49.—Sectional header with one handhole for four tubes.

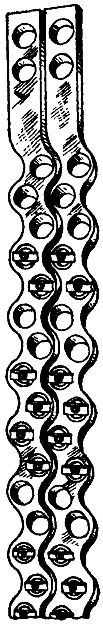


Fig. 50.—Sectional header with round handholes.

the 4-in. diameter in the medium-pressure range. But as pressure increases, metal thickness in both tube and header must increase, which adds to manufacturing expense. Hence to reduce the increase in metal thickness, tube diameter is often decreased for high pressures. Maximum tube length for straight-tube boilers is between 24 and 26 ft., 18 to 20 ft. being about standard.

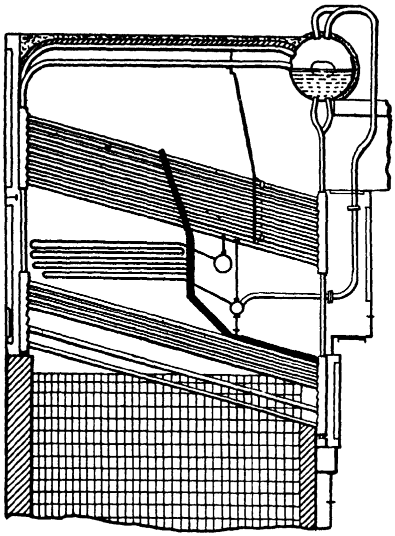


FIG. 51.—Cross-drum straight-tube sectional-header boiler with slag screen to prevent ash slag from clogging entrance to boiler passes. (*Babcock and Wilcox Company.*)

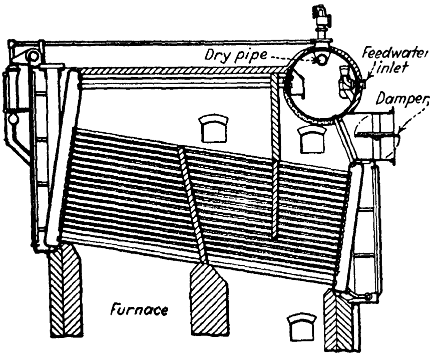


FIG. 52.—Cross-drum straight-tube box-header boiler. (*Combustion Engineering Company.*)

74. Cross-drum straight-tube horizontal-inclined boilers (Figs. 51 and 52) have the same arrangement of straight tubes as the longitudinal-drum types but the drum is placed at right angles to the inclined tubes and above their lower ends. The lower headers, either box or sectional, are connected to the bottom of the drum by short tubes. Headers at the higher end of the tube are connected by a single or double row of tubes to the steam space of the drum. The bottoms of the lower bank of headers are connected by short expanded nipples with a mud-drum header which extends entirely across the boiler. Sludge and sediment concentrates in this mud drum and are drawn off periodically through blowdown valves.

75. Location of the superheater is largely determined by the desired final steam temperature and plays an important role in the general arrangement of the boiler. Ordinarily, higher temperatures dictate the use of a superheater placed between two banks of tubes, as in Fig. 51, and called an *interdeck superheater*. For lower temperatures there may be a choice between a superheater at the top of the first pass or a smaller interdeck superheater. Space required to accommodate superheater surface is so great that it becomes necessary in many instances first to design the superheater, then to consider the boiler. Superheaters are treated in greater detail later.

76. Bent-tube boilers all have one or more top steam drums and a single bottom or mud drum, except for the extremely large boilers used in central stations which sometimes have two mud drums. The drums are connected together by tubes bent so as to enter the drums radially and are expanded into holes drilled and reamed in the drums. When more than one steam drum is used, they are connected together with steam- and water-circulating tubes whose purpose is to permit circulation in the boiler and to equalize pressure and water level by permitting steam and water to flow from one drum to the other. Tubes 4 in. in diameter are not used in bent-tube boilers, largely because of decreased ligament efficiency with increased diameter and the consequent necessary increase in drum-wall thickness. Below

500 lb. per sq. in., $3\frac{1}{4}$-in. tubes are usually used; above this pressure some manufacturers drop to 3-in. tubes. More recently, 2-in. diameter tubes have appeared in certain designs.

77. Bent-tube boilers lend themselves to a greater variety of arrangements than does the straight-tube boiler, and some of these variations are shown in Figs. 53 to 62. They vary all the way from two-drum boilers to four-drum units. Even five, six, and eight drums are used in special high-capacity tailor-made units for central stations and large industrials,

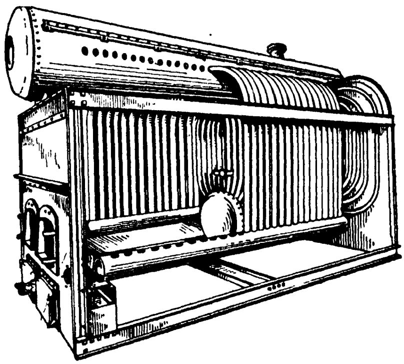


FIG. 53.—Two-drum bent-tube boiler with one drum approximately half length. (*E. Keeler Company.*)

some of which have produced over a million pounds of steam an hour and are shown in outline by Fig. 62.

78. Two-drum bent-tube boilers (Figs. 53, 54, and 55) consist of single upper and lower parallel drums, connected by bent tubes that may be either inclined or practically vertical. The most recent form of two-drum boilers is illustrated in Fig. 55. Here the completely water-cooled furnace is an integral part of the boiler, and the boiler is engineered from the furnace requirements. Both $3\frac{1}{4}$- and 2-in. diameter tubes are used. The larger tubes are used in the first pass, because more steam is produced there and hence a larger tube area is necessary both to take care of circulation and the greater likelihood of scale. Spacing between the tubes in the first few rows is greater so as to prevent slag from accumulating

on the tubes in such a way as to clog the gas passages. Two-inch tubes are used in the second and third passages because more heating surface can be obtained in a given space and heat transfer per square foot is improved by the closer tube spacing. Less steam is generated in these sections of the boiler, hence less scale forms and the tubes can safely be made

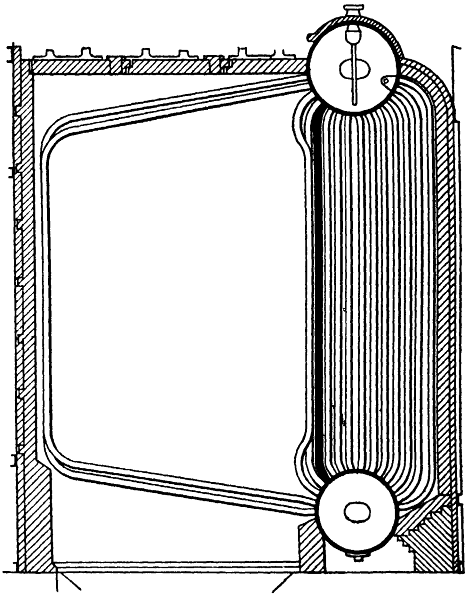


FIG. 54.—Two-drum vertical-bent-tube boiler. Gas flows in a horizontal plane toward the back of the boiler and then forward through the tube bank. (*Steam Generator Company.*)

smaller without danger of the impeding circulation of boiler water.

79. Baffling in this integral boiler differs from that usually employed in that the gas flows nearly parallel to the drums, whereas the usual baffling forces the gas to zigzag through the tube banks vertically. For about half the width of the boiler, the first row of tubes is baffled for their entire length, and the furnace gas is forced to travel to the rear of the furnace before entering the bank of tubes. Vertical baffles

then force the gas to flow towards the front of the boiler in three passes, leaving the boiler either at the top or bottom as most convenient.

80. Superheater is placed in the first pass immediately behind the bank of $3\frac{1}{4}$-in. tubes. It is of the hairpin type and is fed from the steam drum by a number of tubes rolled

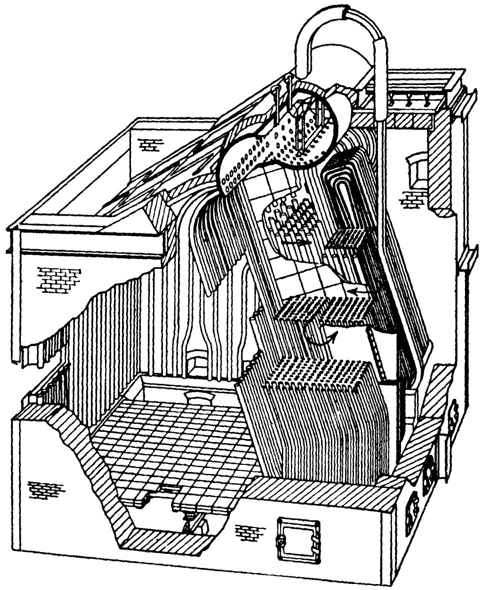


FIG. 55.—Two-drum inclined-bent-tube boiler with integral furnace. Gas flows in a horizontal plane. Both $3\frac{1}{2}$- and 2-in. diameter tubes used. (*Babcock and Wilcox Company.*)

into the drum and the superheater inlet header. This arrangement helps to distribute steam throughout the length of the superheater header and is used in many types of boilers.

81. Waterwalls which protect the furnace consist of tubes bent to conform to the shape of the furnace with their upper ends rolled into the steam drum and their lower ends rolled into headers which are supplied with boiler water from the bottom drum.

82. The three-drum bent-tube boiler has two steam drums and one bottom drum. The drums are connected with tubes

as shown in Fig. 56. Feed water is usually fed to the rear drum and circulates down the back tubes. Steam generation is most vigorous in the front rows of tubes, and this forces an upward circulation of the boiler water. Steam liberated from the water in the front drum passes through the top circulating tubes, which connect the two steam drums, into

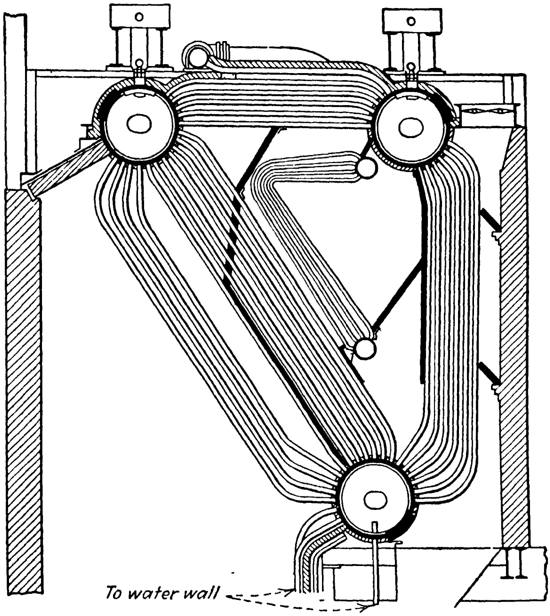


Fig. 56.—Three-drum bent-tube boiler with rear wall water cooled. (*Erie City Iron Works.*)

the rear steam drum. Saturated steam is delivered from the rear drum either to a superheater or to the plant. This type of boiler is used for steam pressures up to as high as 1,400 to 1,800 lb. per sq. in. and in capacities of 25,000 to 700,000 lb. of steam an hour.

83. The low-head boiler of Fig. 57 is a modification of the three-drum boiler developed to permit the modernization of old boiler rooms where headroom is limited. By decreasing the headroom required by the boiler more adequate furnaces

can be constructed in a given available heighth. These boilers may be operated up to about 250 per cent rating, they are rather difficult to baffle, and the heating surface per foot

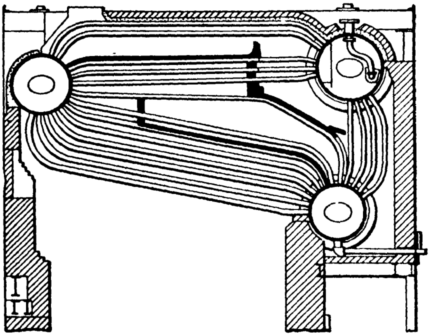


FIG. 57.—Three-drum bent-tube low-head boiler. (*Henry Vogt Machine Company.*)

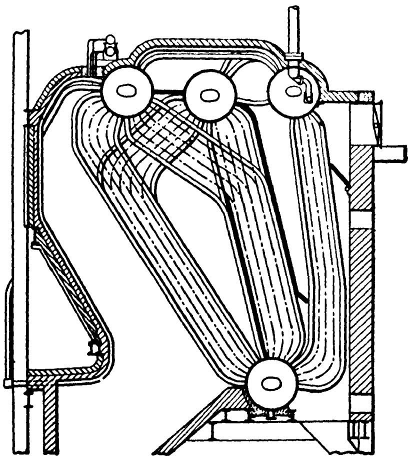


FIG. 58.—Four-drum bent-tube boiler with alternate tubes of first and second banks crossed. (*Combustion Engineering Company.*)

width is low. They are usually used where capacities of not over about 30,000 lb. of steam per hour is required.

84. Four-drum bent-tube boilers (Stirling type) are shown in Figs. 58, 59, 60 and 61, which show most of the principal

variations of this type boiler. They are used when the pressure is less than 900 lb., the three-drum type being used for higher pressures principally because of the high cost of drums for high pressures. Considerable variation exists in the elevations at which the three upper or steam drums are placed. In some designs, all steam drums are at the same or nearly the same level, whereas in others, one of the upper drums

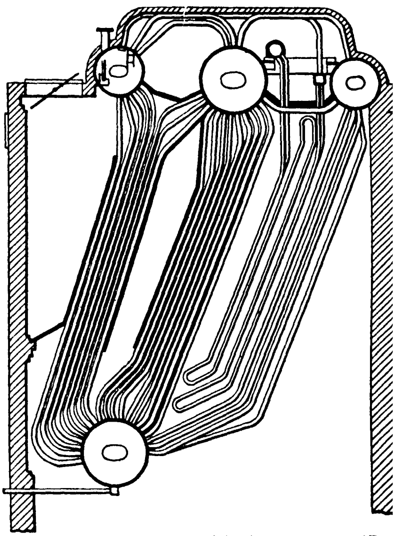


FIG. 59.—Four-drum bent-tube boiler with slag screen. (*Babcock and Wilcox Company.*)

operates completely submerged. Boilers of both types give satisfactory performance.

85. Steam is delivered sometimes from the rear drum and sometimes from the middle drum depending upon the manufacturers idea of which will result in delivery of driest steam. As in the three-drum type boiler, feed is fed to the rear drum. In some boilers baffles shown dotted in Fig. 60 are included in the back steam drum and mud drum. These baffles force feed water to circulate down in the four back rows of tubes, up in the next two rows, and down in the remaining row. This not only gives an economizer effect but also

provides positive circulation of water in the last row of tubes where circulation usually is lazy.

86. Drum diameters vary from 34 to 54 in. depending upon the capacity and the water surface necessary to obtain satisfactory steam liberation. In Fig. 58, alternate tubes in the front bank are rolled into the middle drum, and alternate tubes of middle bank are rolled into the front drum. Since

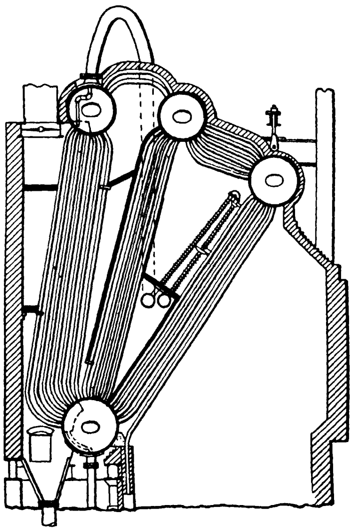


FIG. 60.—Four-drum bent-tube boiler with upper front drum submerged and finned tube superheater. (*Foster Wheeler Corporation.*)

the front bank generates considerably more steam than the middle bank, steam released in the two drums is more evenly distributed, permitting smaller diameter drums or better steam separation. Likewise the ligaments between tubes are larger, permitting thinner drums. Also the steam circulating tubes from the front drum lead steam to the back drum direct, helping to equalize drum pressures and to maintain water level. In Fig. 61, the top front drum acts as a steam-separating or drying drum and is not a part of the boiler circulation. Superheater tubes are rolled directly into this drum.

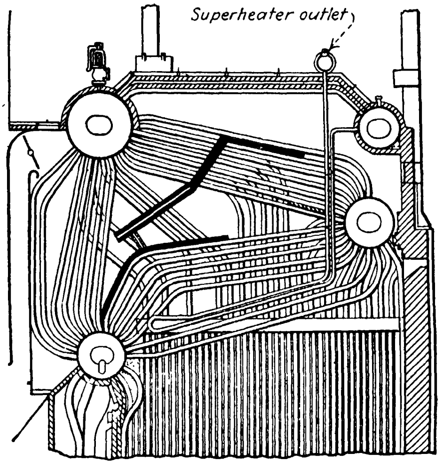


FIG. 61.—Four-drum bent-tube boiler with front upper drum used as a steam dryer. (*Riley Stoker Corporation.*)

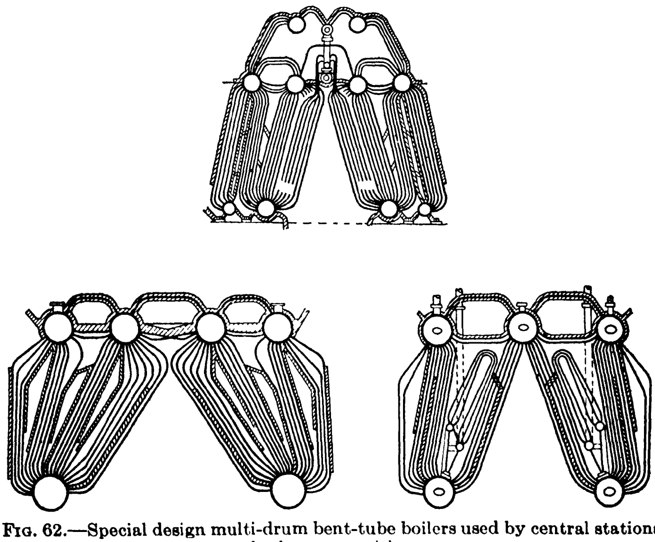


FIG. 62.—Special design multi-drum bent-tube boilers used by central stations for large capacities.

TABLE II.—POWER BOILERS—TYPES, MANUFACTURERS

Maximum figures for usual and occasional practice†	General type of boiler*																	
	Fire-tube								Water-tube									
	Horizontal return tubular	Firebox H.R.T.	Scotch marine	Locomotive (stationary)	Vertical	Special design horizontal, internally fired	Special design horizontal, externally fired	Horizontal, straight, tube longitudinal drum	Horizontal, straight tube cross drum	Vertical straight tube, 2 drum	Bent tube, 2 drum	Bent tube, 3 drum	Bent tube, 4 drum	Inclined straight tube	Bent tube, 3 drum express	Bent tube, 4 drum with radiant element	Bent tube, self contained unit, two or more drums	Double set bent tube with five or more drums‡
Water heating surface, sq. ft.																		
Usual maximum	2,000	2,000	3,000	2,000	1,500	......	......	8,000	20,000	5,000	......	10,000	15,000	6,000	8,000	5,000	10,000	
Occasional maximum	4,000	3,000	3,500	2,500	3,500	......	......	15,000	40,000§	6,000	......	45,000	25,000	30,000	12,000	15,000	15,000	
Capacity, lb. steam per hr.																		
Usual maximum	8,000	6,000	10,000	7,000	5,000	....	....	100,000	300,000	.	.	150,000	200,000	25,000	100,000	50,000	100,000	
Occasional maximum	20,000	10,000	15,000	13,000	12,000	....	....	150,000	600,000§	...	150,000	700,000	450,000	150,000	150,000	200,000	150,000	1,250,000
Operating pressure lb. per sq. in.																		
Usual maximum	150	150	200	200	150	..	..	350	750	250	250	400	450	250	400	250	400	
Occasional maximum	350	200	250	350	200	..	..	450	1,400§	.	600	1,400	900	500	650	......	800	1,400
Manufactures																		
Almy Water Tube Boiler Co., Providence, R. I.								X	X									
Ames Iron Works, Oswego, N. Y.	X	X		X	X													
Babcock & Wilcox Co., New York, N. Y.								X	X			X	X		X		X	X
Bass Foundry & Machine Co., Fort Wayne, Ind.	X							X	X	X		X	X					
Bigelow Co., New Haven, Conn	X	X	X		X							X	X	X				
Broderick Co., Munice, Ind				X														

Wm. Bros Boiler & Mfg. Co., Minneapolis, Minn.	X	X	X	X	X							X	X				X	
Brownell Co., Dayton, Ohio	X	X	X	X	X													
Combustion Engineering Co., New York, N. Y.	X		X		X		X	X	X		X	X‖	X		X		X	X
Erie City Iron Works, Erie, Pa.	X	X		X	X			X	X		X	X‖	X					
Farrar & Trefts, Buffalo, N. Y.	X	X	X	X	X													
Fitzgibbons Boiler Co., Inc., New York, N. Y.	X			X														
Foster Wheeler Corp., New York, N. Y.									X		X	X‖	X		X	X	X	X
Gem City Boiler Co., Dayton, Ohio			X															
Hamburg Boiler Works, Hamburg, Berks Co., Pa.					X													
Hodge Boiler Works, E. Boston, Mass.	X	X	X	X	X													
International Boiler Works, E. Stroudsburg, Pa.	X	X	X	X	X	X	X	X						X				
International Engr. Works, Framingham, Mass.	X	X	X	X	X				X	X	X	X	X					
Johnston Bros. Inc., Ferrysburg, Mich.	X	X	X	X	X	X												
E. Keeler Co., Williamsport, Pa.	X		X					X	X			X‖	X				X	
Kewanee Boiler Corp., Kewanee, Ill.	X	X		X														
Kroeschell Boiler Co., Chicago, Ill.	X	X	X	X	X	X		X	X		X	X	X				X	
Lasker Boiler & Engr. Co., Chicago, Ill.	X	X	X	X	X												X	
James Leffel & Co., Springfield, Ohio	X		X															
Murray Iron Works, Burlington, Iowa	X	X	X	X	X	X	X	X	X		X	X‖	X	X				

TABLE II.—POWER BOILERS—TYPES, MANUFACTURERS.—(*Continued*)

Maximum figures for usual and occasional practice†	General type of boiler*																	
	Fire-tube								Water-tube									
	Horizontal return tubular	Firebox H.R.T.	Scotch marine	Locomotive (stationary)	Vertical	Special design horizontal, internally fired	Special design horizontal, externally fired	Horizontal, straight, tube longitudinal drum	Horizontal, straight tube cross drum	Vertical straight tube, 2 drum	Bent tube, 2 drum	Bent tube, 3 drum	Bent, tube, 4 drum	Inclined straight tube	Bent tube, 3 drum express	Bent tube, 4 drum with radiant element	Bent tube, self contained unit, two or more drums	Double set bent tube with five or more drums‡
Pierce, Butler & Pierce Mfg. Co., Syracuse, N. Y.	X	X		X	X													
Riley Stoker Corp., Worcester, Mass.											X	X‖	X				X	
Springfield Boiler Co., Springfield, Ill.									X			X‖	X		X			
Stanwood Corp., Cincinnati, Ohio	X	X		X														
Steam Generator Corp., Chicago, Ill.																	X	
Titusville Iron Works Co., Titusville, Pa.	X	X	X	X	X	X	X	X	X		X	X	X	X				
Tudor Boiler Mfg. Co., Cincinnati, Ohio	X				X			X						X				
Union Iron Works, Erie, Pa.	X	X	X	X	X		X	X	X		X	X‖	X			X	X	
Henry Vogt Machine Co., Louisville, Ky.	X		X		X			X	X		X	X‖	X					
Wickes Boiler Co., Saginaw, Mich.	X		X					X	X	X		X‖	X		X		X	

* This table includes only boilers which the manufacturer reports he can supply in sizes above 1,000 sq. ft. and pressures above 15 lb.
† Complete consistency is impossible in this data section.
It should not be inferred that all manufacturers can furnish the indicated usual maximum size or pressure.
In large units, particularly those with large radiant surface and economizers and air heaters, the heating surface figures are of little significance.
‡ Made in large sizes only.
§ These limits apply to sectional header boilers only.
‖ Including low-head types.

87. Dry steam is one of the requisites of a good boiler even though equipped with a superheater. One of the early devices used to secure delivery of dry steam was the simple "dry pipe," still used in many boilers. It consists of a pipe of about the same diameter as the boiler outlet nozzle, placed close to the top of the drum. The top side of the pipe is perforated, and steam rising from the water in the drum makes a 180-deg. turn to enter the holes. As long as ratings are not too high, this keeps water entrained by the steam from entering the outlet nozzle. With higher ratings, more elaborate arrangements are often necessary.

88. Dry pipe arrangements in the drums of straight- and bent-tube boilers are shown in Figs. 63 to 66. Figure 63 shows the application in the drum of a longitudinal boiler. A baffle plate is placed over the box-type riser so as to prevent steam as

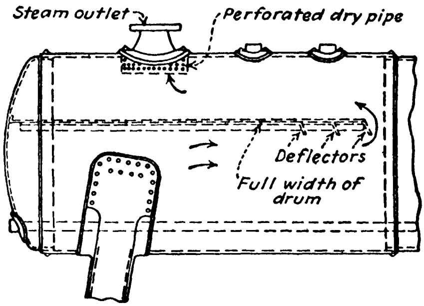


FIG. 63.—Baffle and dry pipe in a box-header longitudinal-drum boiler.

it leaves the riser from splashing water against the dry pipe. Figures 64 and 65 show dry pipes in straight- and bent-tube boilers. Baffle plates prevent steam and water from the circulators splashing against the dry pipe. They also help to throw out water by forcing the steam to make an additional turn before entering the dry pipe.

89. A dry pan takes the place of the dry pipe in Fig. 66 and shutter baffles are introduced to help throw moisture out of the steam. Steam enters the space between the dry pan and the drum wall through perforations. The dry pan does not extend the full length of the drum, but extends over groups of tubes that supply the superheater.

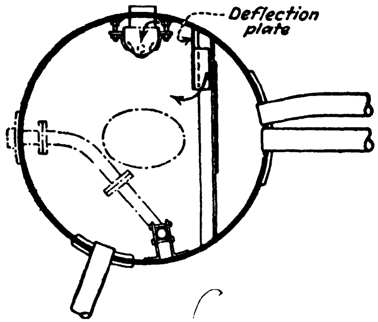


FIG. 64.—Dry pipe and baffle in a cross-drum boiler. (*Combustion Engineering Company.*)

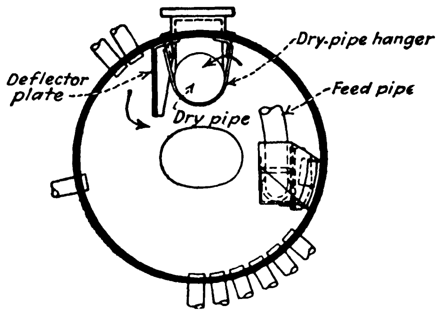


FIG. 65.—Dry pipe and baffle in bent-tube boilers. (*Combustion Fngineering Company.*)

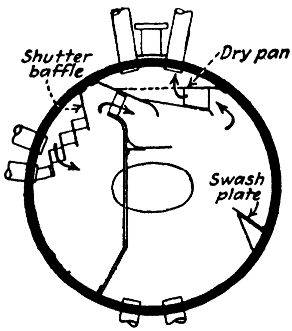


FIG. 66.—Shutter baffles and dry pan. (*Babcox and Wilcox Company.*)

90. Vertical Baffles that extend from just above the steam circulators to the bottom of the drum are sometimes used instead of either dry pan or dry pipe. Steam discharged from the circulators is forced to travel to the ends of the drum

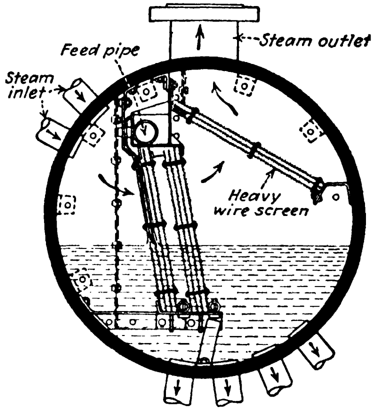


FIG. 67.—Steam washer with screen dryers. (*Combustion Engineering Company.*)

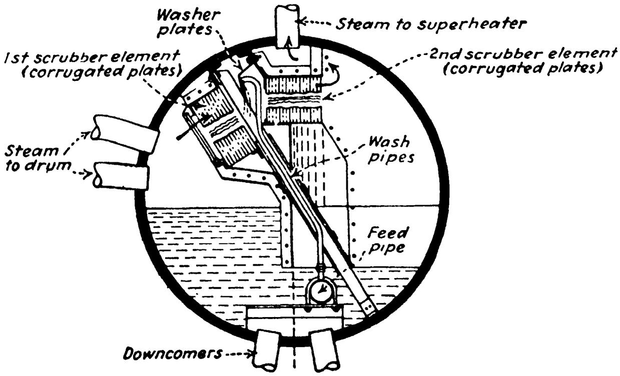


FIG. 68.—Steam washer with baffles arranged to dry steam before washing. (*Babcock and Wilcox Company.*)

at low velocity, where it flows around the ends of the vertical baffle to the main steam space.

91. Deposits of solids on the low-pressure blading of turbines, supplied with steam from boilers operated at high

ratings and high concentrations, have been experienced in a number of high-pressure installations. Such deposits lower the capacity of the turbine and decrease its efficiency. Steam washers such as are shown in Figs. 67 and 68 have been developed. Steam is first washed by entering feed water which has a lower solid concentration than the water in the boiler and hence dilutes the concentration of solids in water entrained with the steam. The washed steam is then dried by corrugated plates or heavy screening before passing to the steam outlets. The washer of Fig. 68 dries the steam before it is washed. Steam washing is particularly necessary when the boiler water contains silica, as silica carry-over is very difficult to remove from turbine blading. Steam washers reduce the solids in the delivered steam to 1 p.p.m. and moisture to ¼ per cent.

QUESTIONS ON DIVISION 5

1. How does a water-tube boiler differ from a fire-tube boiler?

2. Name and describe four types of straight-tube water-tube boilers.

3. What is the difference between box and sectional headers?

4. In straight-tube boilers where may the superheater be located?

5. Above what pressure do boxheader boilers become uneconomical?

6. How many drums have bent-tube boilers?

7. How are bent-tube boilers supported?

8. What diameter tubes are used in bent-tube boilers? In straight-tube boilers?

9. Describe the Stirling type boiler.

10. In a three-drum bent-tube boiler where is the feed water usually introduced?

11. What is a dry pipe? What is it used for? What other devices are used to prevent delivery of wet steam?

12. What is a steam washer? Describe how one type functions?

DIVISION 6

SUPERHEATERS

92. The superheater takes dry steam from the boiler and raises its temperature well above the saturation temperature corresponding to the steam pressure. Final temperatures as high as 1000° have been used and temperatures of 850 to 950° are being adopted for new central stations. Superheaters take many forms and shapes because of the great variety of

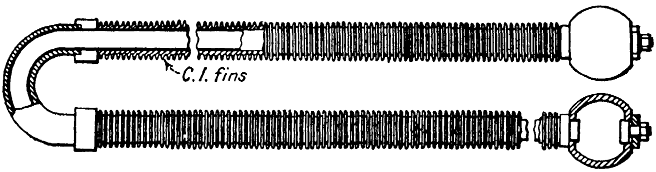


FIG. 69.—Foster Wheeler fin-tube superheater element.

boilers and irregularity of the space available in the boiler for its location.

93. Steam is superheated by passing it through tubing which receives heat in a manner similar to that which is observable in other parts of the boiler. The superheating apparatus is called a *superheater*. Superheaters are of two general types: (1) smooth and (2) extended surface, the latter having cast-iron fins or gills shrunk upon a smooth tube as in Fig. 69 to increase the heating surface. They may also be classified as *convection* or *radiant* as determined by location in the boiler. The saturated steam from the boiler passes through the superheater-tube elements, whence in the condition of superheated steam it enters the steam main. Superheater tubes are usually 2 in. in diameter.

NOTE.—A superheater should (Hirshfeld and Barnard, "Heat-Power Engineering") fulfill the following requisites: (1) perfect freedom of expansion; (2) ability to withstand high temperatures, high pressure, and sudden changes of temperature; (3) nonexposure of joints to the hot

gases; (4) access for cleaning externally and internally; (5) means for

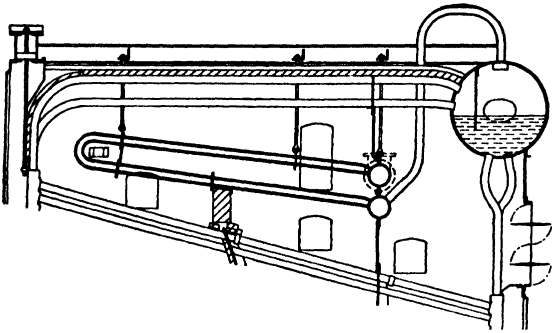


FIG. 70.—Babcox and Wilcox hairpin superheater in overdeck section of a sectional-header boiler.

adjusting the superheat to any desired temperature; (6) automatic control of the desired temperature; (7) provision, in some cases, for flooding the apparatus with water and for draining it; (8) occupation of small space; (9) low first cost; (10) small expense for operation and maintenance; (11) low-pressure drop.

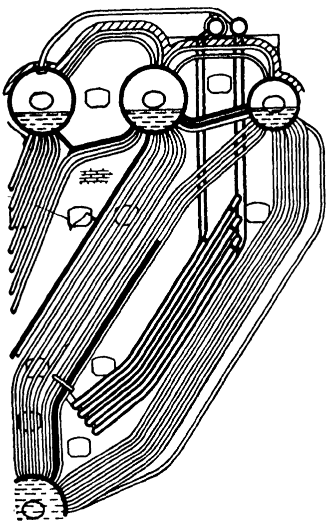


FIG. 71.—Pendant type superheater in three-pass four-drum bent-tube boiler.

94. A smooth-tube hairpin type superheater is shown in Fig. 70 placed in the overdeck section of a sectional-header cross-drum boiler. Flow to steam in the saturated header is in the opposite direction to superheated steam in the discharge header to insure equal-pressure drop across all tube elements. To further insure equal distribution of steam to the super-heater tubes so as to prevent starving and consequent overheating, saturated steam is taken from the boiler drum by a number of tubes rolled into the drum and into the header. Sometimes the superheater tubes are themselves rolled into the steam drum as in Figs. 59 and 75.

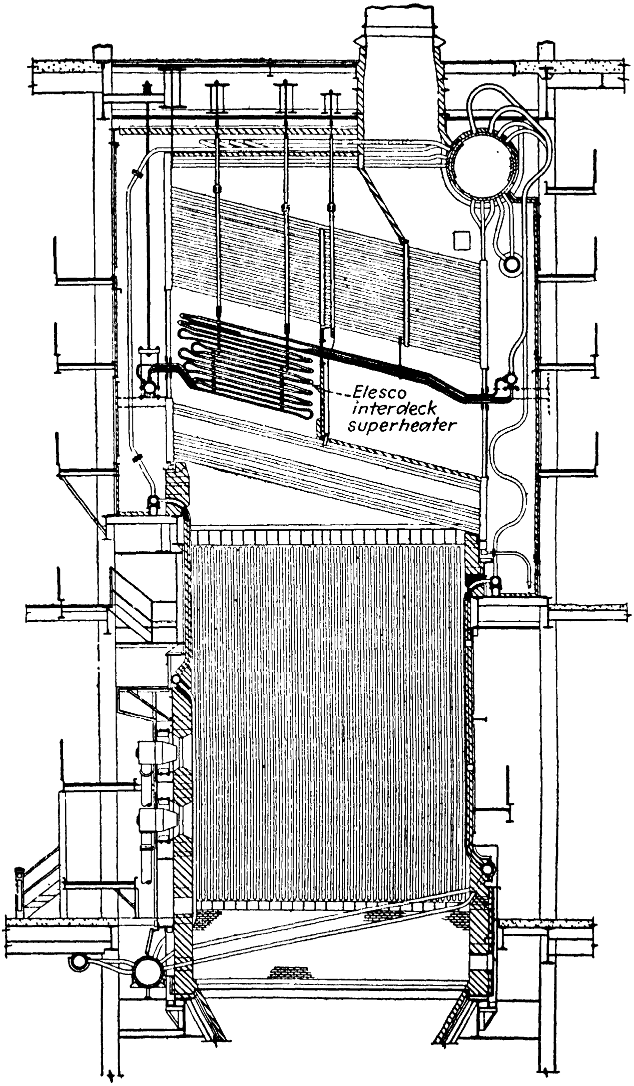


FIG. 72.—Multi-loop inter-deck superheater with headers outside of boiler casing. (*Superheater Company.*)

95. Multiple-loop superheaters consist of many elements, each a continuous tube bent back on itself a number of times between an inlet and outlet header. Ordinarily each tube is bent to lie in a single plane and is inserted vertically between tubes or in an open space between tube banks. Figure 71 shows a multiple-loop superheater in a Stirling boiler, and Fig. 72 shows a multiple-loop interdeck superheater.

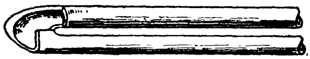


FIG. 73.—Forged return bend used in Elesco superheaters. (*Superheater Company.*)

Modern practice places the superheater headers outside of the boiler, thus making them more accessible and at the same time removing the tube and header connections from the high temperature zone. In Fig. 71 the tube elements are bent at constant radius, and in Fig. 72 the bends at one end are drop-forged in the manner indicated in Fig. 73. The tubes of this type are connected to the headers by detachable metal-to-metal ball joints, secured by heat-treated steel studs and clamps. Any joint can be broken merely by removing the nut that holds the clamp unit as shown by Fig. 74.

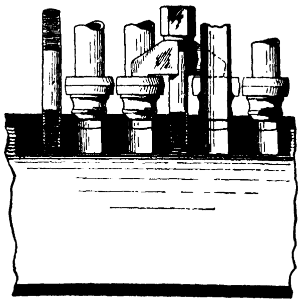


FIG. 74.—Normal pressure Elesco superheater header and ball joints in section, showing how the unit ball ends fit into ground 45-deg. conical seats in the header and are held in place by drop-forged clamps and heat-treated steel studs. Unit ball ends are forged integrally with the tubing.

96. Exact control of superheat becomes necessary as steam temperatures go higher and safety margins become less. Conversely, closer control permits a higher steam temperature with safety. In a superheater heated only by convection, steam temperature rises with load. The reason is obvious. If the load doubles, twice as much steam passes through the superheater, but this is more than offset by higher gas temperatures and doubled gas velocity. The effect of load change is reversed in a superheater heated only by radiant heat. When boiler load is doubled, radiant heat delivered to the

superheater does not double, so steam temperature falls. While a radiant and a convection superheater can be connected in series to produce a practically flat temperature-load curve, it is more common practice to use a single superheater placed to receive part of its heat by radiation and part by convection.

97. Temperature control with convection superheaters may be obtained by various devices. A recent installation connects two superheaters in series, with a desuperheater between.

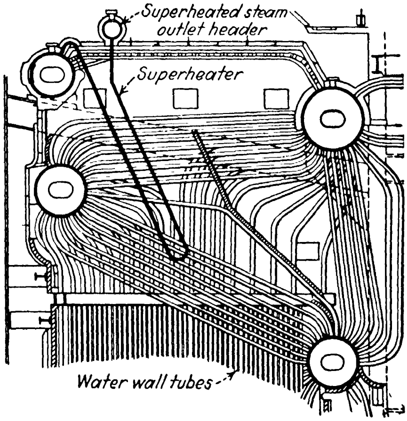


FIG. 75.—Low-temperature superheater between tubes of a Riley boiler.

A thermostatically controlled by-pass around the desuperheater maintains constant temperature at the outlet of the second superheater. One simple type of control involves special baffling and dampers whereby any desired part of the gas flow can be by-passed around the superheater. Still another, preferred by many engineers, is the compensating superheater. In its most common form, this is a single superheater split into two unequal parts by the boiler baffling. Gas flow through the two parts of the superheater traces two paths through the boiler. Dampers apportion the gas flow between the two paths and thus permit exact control of superheat.

Another method of superheat control is shown in Fig. 432 where a vertical baffle in a single-pass boiler, extending from

the tube bank to the boiler gas outlet, divides the upper part of the boiler into two sections. The superheater is placed in one of the sections so formed and a continuous-tube economizer in the other. By controlling the dampers in the outlet from each section the amount of gas passing over the superheater can be regulated and so control accurately the final steam temperature. In starting up the boiler, the dampers on the superheater side are kept closed, thus preventing overheating of the superheater elements which sometimes happens in the usual arrangement because of lack of steam flow to keep the tubes cool.

98. Tube Material.—The limiting temperature of a carbon-steel superheater is a function not only of the desired steam temperature, but also of the unbalance between steam flow and gas flow in various sections. In any sections where inside metal temperature exceeds 950°F., tubes of chrome-nickel or chrome-molybdenum alloys are used. These are available to withstand inside metal temperatures as high as 1200°.

99. Superheated steam is a simple means for obtaining improvement in economy of a steam power plant. It effects improvement in the following four ways:

1. By reducing the steam consumption of the prime mover
2. By reducing condensation losses in steam lines.
3. By eliminating erosion of steam turbine blades.
4. By increasing the capacity of the plant.

100. Steam consumption of turbines is reduced about 1 per cent for each 10° of superheat. This saving is partly due to increase in volume of the steam but mostly because absence of moisture decreases friction losses. Rapid erosion of turbine blades by moisture in steam has been an important factor in forcing the general adoption of superheated steam for turbines. Saving in steam consumption by use of superheated steam in engines is about 1.5 to 2 per cent for each 10° of superheat. It is due to the greater volume of superheated steam and therefore to the less weight of steam required to fill the cylinder up to point of cutoff and to decreased cylinder condensation.

101. Steam reheaters are similar in form and purpose to superheaters. In plants employing pressures of 1,000 lb. per

sq. in. or over, the steam in expanding through the turbine becomes very wet and in this condition will cause damage to the turbine blading. Unless the initial steam temperature is very high, it becomes necessary to bring the steam back to the boiler and pass it through a reheater which drys it and superheats it usually to the original initial temperature. It is then expanded through the low-pressure stages of the turbine.

QUESTIONS ON DIVISION 6

1. What are superheaters used for?
2. What two types of superheaters are in use?
3. In what two ways do superheaters receive heat?
4. What diameter tubes are usually used in superheaters?
5. What requisites should a superheater fulfill?
6. How is the steam delivered to the superheater?
7. Describe two methods used to control superheat?
8. What is a multiple-loop superheater?
9. What is the highest steam temperature in use?
10. Of what materials are superheater tubes made for usual temperatures? For very high temperatures?
11. What are the advantages of superheating steam?

DIVISION 7

WATERWALLS

102. Waterwalls consist of relatively closely spaced vertical or horizontal tubes placed in the furnace walls, supplied with water from the boiler circulation, and delivering the steam generated in them to the steam drum of the boiler. They were deveolped to cool the refractory lining of the furnace so decreasing maintenance and increasing boiler reliability. Though first used with stokers they are now applied to one or all four walls of pulverized-fuel fired, oil fired, and gas fired furnaces. Waterwalls materially decrease the temperature of the furnace gas at entrance to the boiler, thus permitting higher combustion rates and greater heat release per cubic foot of furnace volume before trouble is experienced with ash slagging on the boiler tubes.

103. Heat transfer to waterwalls is mostly by radiation and thus the surface forms a very active part of the boiler. As a square foot of waterwall surface often evaporates as much as 100 lb. of steam an hour it adds considerably to the boiler capacity or in other words decreases the surface necessary in the boiler proper for a given desired output. Though waterwalls decrease furnace temperature, they decrease exit-gas temperature but little more than an equal surface in the main part of the boiler. Because of the high rate at which the surface works, it must be amply supplied with relatively scale-free water, circulation must be rapid, and the tubes arranged so that none of them are starved. Tube burnouts and blisters will be sure to follow if any of these requirements are neglected.

104. Water feed for the walls is taken from the mud drum in the case of bent-tube boilers, or from the main drum in a horizontal straight-tube boiler, through a number of tubes rolled into the drum, usually at the ends. Occasionally the

downcomers are made to project into the mud drum, as in Fig. 76, to avoid danger of sediment being fed to the water-

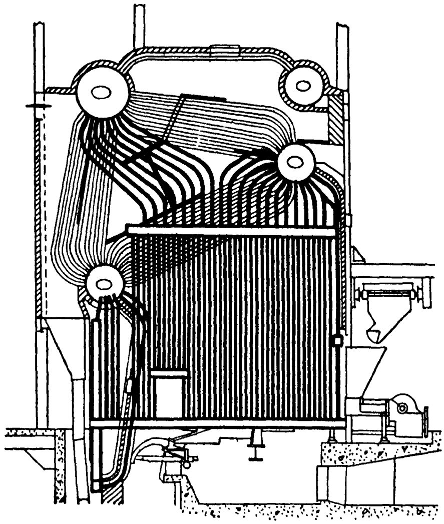


FIG. 76.—Water-cooled furnace with waterwall headers carried in wall and down-comer tubes projecting into mud drum.

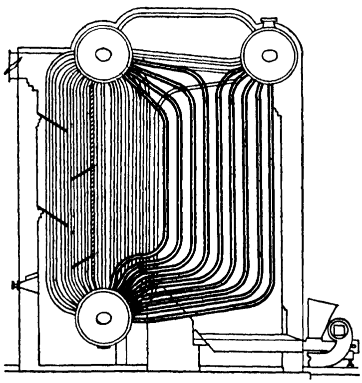


FIG. 77.—Side wall with tubes rolled into mud drum and steam drums.

wall. These tubes feed headers at the bottom of each wall. Sometimes for convenience, particularly in the case of straight-

tube boilers, tubes from the drum are rolled into a stub header connected to a single large-diameter pipe feeding the wall bottom headers. When the downcomer tubes are rolled into the bottom headers at regular intervals throughout their length, better distribution of feed water is secured.

105. Riser tubes rolled into the bottom headers form the waterwall and are usually rolled into a top header, from which

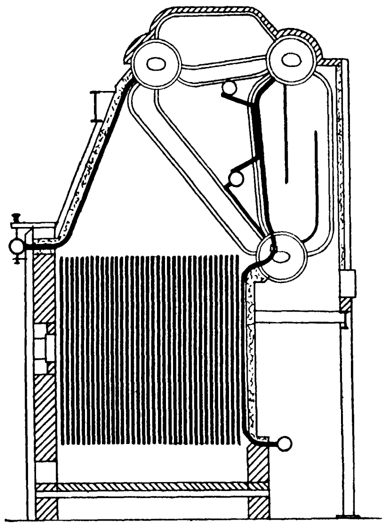


FIG. 78.—Waterwall arrangement showing back-wall tubes carried through mud drum.

a smaller number of tubes deliver the steam and water mixture to the steam space of the boiler drum. An exception to this is shown in Fig. 77, where the side-wall tubes are carried up individually into the end of the steam drum. Another exception is shown in Figs. 76 and 78, where the rear-wall riser tubes are rolled into the mud drum. Sometimes these tubes are carried on through the mud drum and connected to a row of boiler tubes, as in Fig. 78.

106. Tube spacing or arrangement has not been standardized as may be seen from the illustrations. Tube spacing is usually determined by the particular type of protective sur-

facing applied. Yet even when bare tubes are used, spacing may vary from a 3¼-in. diameter tube on 3½-in. centers up to wide spacing, such as in Fig. 77, depending upon how much waterwall surface is required. Either 3 or 3¼-in. tubes are used in waterwalls, depending on pressure. Larger diameter tubes are frequently used for water feed and circulators.

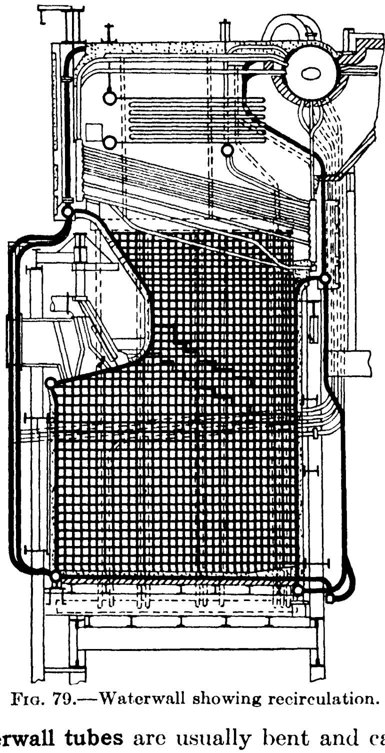


FIG. 79.—Waterwall showing recirculation.

107. Waterwall tubes are usually bent and carried outside of the furnace before they are rolled into the top and bottom headers. Examples of this practice are shown in Figs. 78 and 79. But in one type of wall (Fig. 76) the tubes are straight and rolled into the top or bottom of the headers which are inside the wall.

108. A wall using horizontally inclined tubes instead of vertical is illustrated in Fig. 80. This arrangement is used

particularly for stoker installations when only the wall near the stoker is to be protected. The tubes, inclined so they are parallel with the tuyères, are rolled into headers at each end. The back or lower header is supplied with water from the boiler, and tubes rolled into the front header carry the steam-and-water mixture to the steam drum. With stokers waterwalls are used in the rear and two side walls, but not often in the front wall.

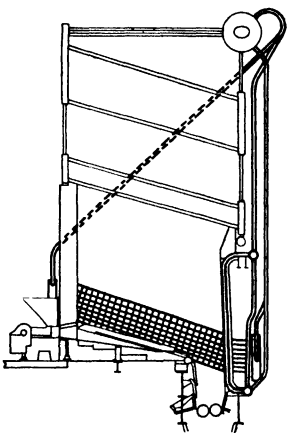


FIG. 80.—Inclined waterwall used with stokers.

109. Recirculator tubes connecting top- and bottom-wall headers outside of the furnace are sometimes provided as in Fig. 79. Since steam is not generated in these tubes, water flows down through them and up in the waterwall tubes, thus creating a circulation. Their use makes it possible to decrease the number or area of tubes connecting the waterwall with the steam-and-water spaces of the main boiler, as these then have only to supply make-up water and carry off the steam.

110. Support of waterwalls is arranged in various ways. In some walls the top-wall header is supported rigidly on steel anchored to building steel or boiler columns. The tubes, lower header, and furnace bottom, if of the slag-tap type, all hang from the upper header, hence are free to expand downward. In some instances the top header is relieved of some of the weight by spring supports under the bottom header. Expansion in the tubes connecting the top header to the boiler drum is usually not great, because the tubes are relatively short. It is taken care of by flexibility designed in the tube connections. Sometimes, particularly with slag-tap furnaces, the bottom headers are anchored, and all expansion is upward. This arrangement prevents movement of the tubes through the furnace floor. The upper header is guided and/or supported on springs and moves vertically upward as the

tubes in the wall expand. In other walls both top and bottom headers are supported on springs. This permits downward movement of both top and bottom headers.

111. Openings through walls for access doors, when the doors are large and tube spacing is close, are made by providing stub headers above and below the opening into which the wall tubes are rolled. If the door opening is small and tube spacing not too close, tubes may be bent so as to go around the opening.

112. Walls made up of bare tubes backed with firebrick or insulating refractory are used in all parts of the furnace except at the fire line and ashpit where protection is usually provided. Some bare-wall tubes have fins welded to them to seal the space between the tubes and present a full metal surface to the furnace. Figure 81 is a section through a waterwall made of bare tubes

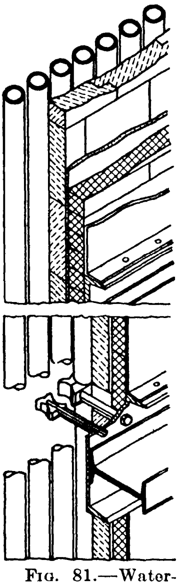


FIG. 81.—Waterwall of bare closely spaced tubes backed with insulation and steel casing. (*Riley Stoker Corporation.*)

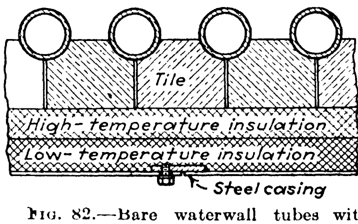


FIG. 82.—Bare waterwall tubes with refractory backing. (*Foster Wheeler Company.*)

closely spaced and shows how the tubes are backed with shiplap tile insulation and steel casing. Special refractory tile shaped to the curvature of the tubes so that the refractory fills in between the tubes as in Fig. 82 are used when the tube spacing is greater.

113. Protection of the waterwall tubes at the fuel line in stoker-fired units and in the lower parts of pulverized-fuel-fired boilers is obtained either with refractory or cast-iron

blocks. Refractory protection decreases the cooling effect of the wall, affording better operation at moderate ratings while at the same time reducing maintenance. One method of protecting with refractory is shown in Fig. 83. The special silicon carbide tile are shaped to fit around the tubes and

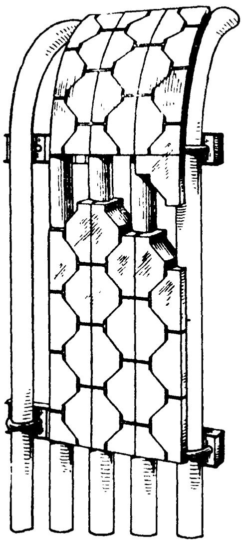


FIG. 83.—Bernitz waterwall protection of silicon-carbide refractory.

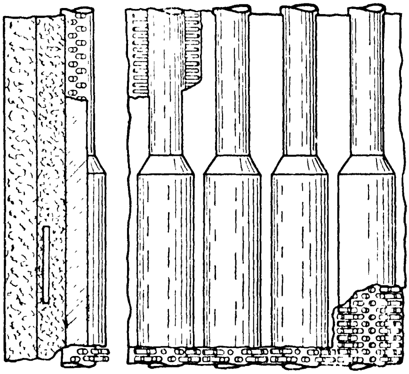


FIG. 84.—Studded waterwall tubes covered with plastic refractory. (*Babcock and Wilcox Company.*)

can be put on the tubes from the inside of the furnace. A method of protecting tubes with refractory used by Babcock

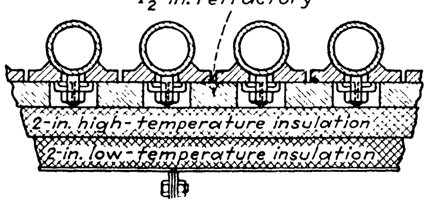


FIG. 85.—Waterwall tubes backed with cast-iron blocks bolted to the tubes, refractory, insulation and steel casing. (*Foster Wheeler Corporation.*)

and Wilcox is shown in Fig. 84. Studs are spot-welded to the tubes on the fire side, and the tubes and studs then covered

with a plastic chrome refractory. The studs hold the refractory in place and also help to keep it cool.

114. Cast-iron block protection is used along the fire line of chain-grate and underfeed stokers to protect the tube from abrasion by the coal and to provide a smooth-plane surface to which clinker will not stick. They are also used to form the floor of slag-tap furnaces and in the lower part of the walls of pulverized-coal furnaces where the flame comes close to the wall. The cast-iron blocks are held onto the tubes by various methods of bolting, as indicated in Fig. 85 and by casting the metal directly on the tube in such a way as to form a fused

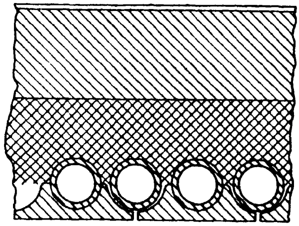


FIG. 86.—Waterwall tubes protected by cast-iron blocks fused onto every other tube. (*Springfield Boiler Company.*)

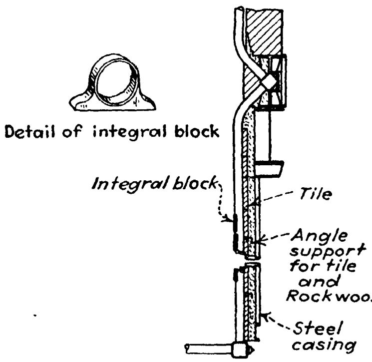


FIG. 87.—Waterwall protected with integral cast-iron block fused to the tubes and backed with rockwool insulation and steel casing. (*Combustion Engineering Company.*)

bond with the tube metal as in the detail of Figs. 86 and 87. In Fig. 88 the tubes are clamped between a tie bar and the cast-iron blocks into which the clamping bolts are tapped. Three types of blocks are used: smooth, rough—to which the coal-ash adheres—and refractory faced. When these blocks are used on the upper part of waterwalls,

they usually are of the rough type so a thin coating of ash will adhere to them or have the refractory embedded in the casting. In Fig. 85 the bolts are spot-welded to the tubes.

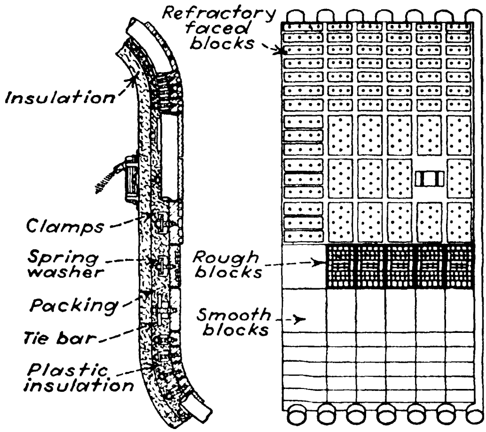


FIG. 88.—Waterwall tubes with smooth, rough and refractory-covered cast-iron protecting blocks. (*Babcock and Wilcox Company.*)

QUESTIONS ON DIVISION 7

1. What are waterwalls used for?
2. What do waterwalls consist of?
3. What rates of evaporation are obtained in waterwalls?
4. By what method do waterwalls receive heat?
5. How are waterwalls tied into the boiler circulation?
6. What are recirculating tubes? What are they used for?
7. Name three materials used to protect tube surfaces?
8. How is metal protection fastened to the tubes?
9. How is refractory fastened to the tubes?
10. How are waterwalls supported?

DIVISION 8

MATERIALS USED IN STEAM-BOILER CONSTRUCTION

115. The materials used in boilermaking are the following: (1) wrought iron, (2) mild or low-carbon steel (see note below), (3) cast iron, (4) cast steel, (5) malleable iron, (6) copper, (7) bronze, (8) brass. This enumeration includes all of the metals that have been employed more or less extensively in the construction of boilers since the pioneer days of steam power.

NOTE.—*Flange*, *firebox*, *rivet*, *stay bolt*, *bar*, and *boiler tube* are terms referring to different grades of mild steel which are specified in the A.S.M.E. Code for boiler construction. *Wrought steel* is any steel that has been *worked* in the process of manufacture. The term is used in contrast to *cast steel*.

116. Quality of the materials used in steam-boiler construction demands critical attention for the following reason: In performing its function (Div. 1) a steam boiler is subjected continually to disruptive stresses. These are due to high internal pressures and to excessive changes in temperature. Disastrous consequences will inevitably follow if the material fails under these stresses.

NOTE.—The steam boiler is exposed to a greater variety of conditions which tend to affect its deterioration than is any other power-plant member. While a boiler is in service, the fibers of its material are subjected to constantly varying molecular stresses due to the continually changing temperatures. This inconstant stress tends to induce a change of molecular structure (crystallization) which diminishes insidiously the strength of the material. Furthermore, the material is subjected often to unnecessary torture and abuse because of accidental circumstances. These may be an excessive steam pressure or a deficiency of water.

117. The quality of the material in the different parts of a boiler should be selected with special reference to the stresses and disruptive influences which each part encounters in service. Certain portions of the structure must be par-

ticularly capable of withstanding pulling stresses. Still others have shearing and crushing forces imposed upon them. One part must have a peculiar hardness to resist the wasting away of the material by erosion. Another must have superior toughness. Still another must be especially adapted to resist the attacks of corroding elements.

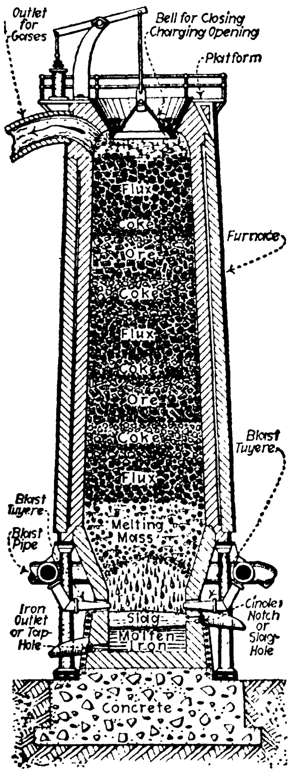


FIG. 89.—Blast furnace.

118. Charcoal Iron Is a High Grade of Wrought Iron.—It contains the highest percentage of pure iron of any of the commercial products evolved from the native ore. It may be used in steam boilers for tubes. It is manufactured from iron ore. There are two stages in the process. (1) The charge of iron ore is smelted into pig iron in a blast furnace (Fig. 89). Wood charcoal alone is used for fuel in making charcoal pig iron. (In making ordinary iron, coke is used). This wood-charcoal is practically free from the impurities which are present in coke and similar fuels and which would affect the purity of the product. (2) The pig iron is melted and rabbled in a puddling furnace (Fig. 90) with oxide of iron. The function of the puddling furnace (Fig. 90) is to remove the carbon from the pig iron. Usually the product of the puddling furnace contains not more than 0.5 per cent (0.005) of impurities. The iron oxide is usually red oxide of iron or hematite ore. The carbon in the pig-iron charge is expelled by its union with oxygen. The two elements (the carbon from the pig iron and the oxygen from the oxide) combine to form carbon dioxide (CO_2) and pass off as such. This converts

the original brittle charge to the malleable state which is characteristic of wrought iron.

119. Wrought iron comes from a puddling furnace as a soft plastic ball saturated with slag. The ball is dropped into a machine (Fig. 91) which squeezes out most of the slag. The "shingled" mass of metal is then passed through a train of rolls (Fig. 92) which ejects much of the remaining slag. During this operation, the plastic mass solidifies into the form of a bar. This "muck-bar" is now cut into strips. A sufficient number of strips to produce a sheet of the desired size are bound into a bundle or pile (Fig. 93). After being brought to a welding heat, the pile is rolled out into a plate of the required thickness.

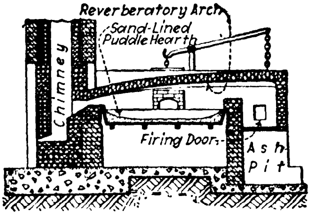


FIG. 90.—Longitudinal sectional elevation of puddling furnace.

120. A Wrought-iron Plate Consists of a Series of Welds (Fig. 94).—This is evidenced by its laminar structure. Its

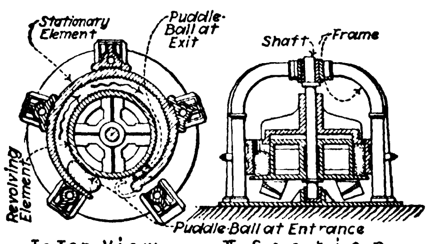


FIG. 91.—Rotary squeezer for squeezing out slag.

fibrous texture is due mostly to the presence of slag in the material. The rolls draw the metal out into a stringy mass. Each fiber of iron is the core of a slender thread of slag.

121. Mild or low-carbon steel is an alloy of pure iron with small proportions of carbon in chemical combination as iron carbides and of other elements. It is manufactured from pig iron. The charge of iron is melted in a reverberatory furnace (Fig. 95). Sufficient oxide of iron is added to insure

the expulsion, in the furnace, of the carbon which was in the original charge. Then the quantity of combined carbon nec-

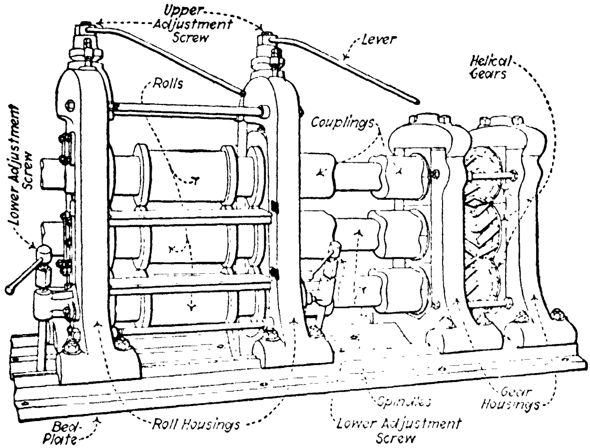


FIG. 92.—Train of rolls.

essary to render the metal structurally adaptable is added. Mild steel made by the Bessemer process is not used for boiler plate.

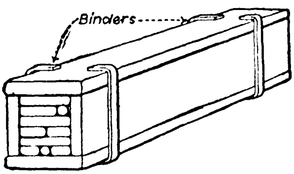


FIG. 93.—Pile of muck-bar strips and wrought iron scrap prepared for reheating.

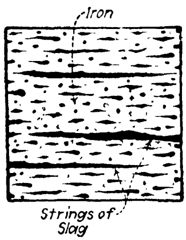


FIG. 94.—Showing lamina, in wrought iron, of slag and iron (magnified 50 diam.).

122. Mild steel for boilers is made by the open-hearth process (Fig. 96). In this process either of two methods—"the basic method" or the "acid method"—may be used. These terms refer to the chemical reactions which occur in the

furnace. Refractory substances, of either an alkaline or an acid nature in the lining of the furnace and in the slag, are the determining factors in the reactions. The slag is the semi-fluid which forms in the furnace from the mixing of the chemical impurities in the metal charge with the fluxing material which is used.

123. The acid is simpler than the basic method but it removes none of the phosphorus and sulphur from the charge of pig iron. The quantity of these impurities is, therefore, greater in the product of the acid method. Hence the basic method is preferred for the manufacture of mild steel. Mild

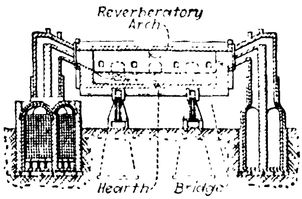


FIG. 95.

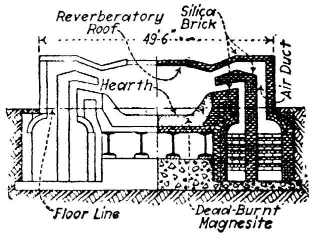


FIG. 96.

FIG. 95.—Reverberatory furnace for steel manufacture. The furnace shown is carried on a track so that it can be moved longitudinally. In a reverberatory furnace the metal is exposed to the action of the flame, but is not in contact with the burning fuel. The flame passes over the bridge, strikes the roof of the furnace and then reverberates, or rebounds, downward on the metal which is spread on the hearth.

FIG. 96. Sectional elevation of basic open-hearth furnace.

steel comes from the furnace in a molten state and is cast into ingot molds. The ingots are then reheated and rolled into plates.

124. Mild Steel Is Distinguishable from Wrought Iron Only in Its Physical Characteristics.—Chemically, low-carbon steel—that is, mild steel—and charcoal or other wrought iron are virtually the same. The principal difference between the two metals is a structural one. The mild steel is homogeneous, whereas the wrought iron is laminated, owing to the different method of its manufacture, and of fibrous texture. Since mild steel has but a trace of combined carbon and other strengthening elements, it is often called *ingot* iron. True or tool steels can be tempered; mild steel can not.

125. Wrought iron has been superseded as a material for boiler plates by mild steel within the last forty years. Mild steel, made by the open-hearth process, is now the only material specified for shells, drums, and fireboxes or any plates that require staying or flanging in boilers.

TABLE III.—ESSENTIAL PHYSICAL PROPERTIES OF MATERIALS FOR BOILERMAKING

Quality	Definition	Criteria	Example
Tenacity . .	Ability to resist a pulling stress	Ultimate tensile strength as determined by tension test	Resistance of through stays
Elasticity	Capacity for resuming normal shape after deformation	Extent of deformation from which specimen will completely recover	Action of spring tubes in pressure gages
Hardness ...	Ability to resist erosion or wear	Behavior of specimen in abrasion test, scratch test, or ball test	Imperviousness of good firebox steel to cutting action of cinders
Ductility ...	Ability to endure elongation without fracture of rupture	Degree of elongation of specimen in tension test	Stretching of stay bolts or plates in tension
Malleability. .	Ability to endure change of shape by hammering, bending, or rolling	Behavior of specimen in bending test	Flow of rivet material under blows of hammer or pressure of riveting machine
Toughness	Ability to endure continued torture by twisting and bending; absence of brittleness	Behavior of specimen in torsion test	Resistance of boiler plate to an alternating flexure in riveted seams
Resilience . .	Capacity for storage of returnable work energy as the material is strained to the elastic limit	Modulus of resilience as determined by tension test	Action of steel spring under tension or compression
Homogeneity	Continuity and uniformity in the grain or fiber of the material	Appearance of fractured surfaces of broken specimen	Condition of a soft-steel rivet after being driven under 80 tons pressure while at a reddish white heat

126. The suitability of manufactured iron or steel for boilermaking is determined by tests—some chemical, some physical. The standard tests are specified in detail in the A.S.M.E. Boiler Code. A chemical test or analysis determines the relative proportions of the various component chemical

elements which are enumerated in the following section. These elements are usually inseparable from the finished product. Excepting carbon, without which iron would be of little or no commercial value, these elements are present in mild steel, principally because it would be prohibitively expensive to eliminate them. A physical test reveals the effects of these elements upon the general strength and durability of the material.

127. The principal elements besides pure iron which compose mild steel as it is used for boilermaking are (1) carbon, (2) phosphorus, (3) sulphur, (4) silicon, (5) manganese.

128. An essential property of boiler plate is a uniform blending of the physical properties that will enable the material to recover from the strains induced by the various stresses of operation.

129. An important property of boiler plate is tenacity or tensile strength. Carbon is the ingredient which enhances this property. Carbon possesses no great strength on its own account, but when it is combined chemically with iron, it then develops greater strength therein. However, to insure this, correct proportions must be maintained. Increasing the carbon content up to a certain maximum augments the strength. But beyond this maximum, the strength decreases with the increase of carbon content.

Example.—Mild steel that contains 0.1 per cent of carbon has a tensile strength of about 50,000 lb. per sq. in. With twelve times this quantity or 1.2 per cent of carbon, the tenacity, if tempered, is increased to nearly 140,000 lb. per sq. in., which is probably the upper limit for carbon steel. Increasing the percentage of carbon above this value results in a proportionate drop in the tenacity. With 2.0 per cent, its unit strength is about 90,000 lb. Further gradual increase in the carbon component causes the material to become brittle.

130. Carbon contributes to the hardness of boiler plate. The hardness increases with the increase of carbon content. This quality is especially desirable in flues and tubes and in the sheets of fireboxes and combustion chambers. In these locations the metal must withstand the abrading action of the cinder-laden gas currents. There is, however, a degree of hardness which marks the maximum limit. If an attempt

is made to obtain harder metal, other very necessary qualities of good boiler plate will be sacrificed.

131. Excessive carbon tends to destroy ductility of the material. Its malleability may also be thereby impaired to a ruinous extent. Likewise, a plate containing an excess of carbon will be lacking in toughness. Sufficient carbon to make the plate quite hard will also make it brittle.

132. Good boiler-plate steel contains just enough carbon to insure proper melting in the furnace. Generally, the quantity of carbon is less than 0.25 per cent. With this small carbon content, practically all liability of the material to harden and crack under a stress, which is caused by a sudden and wide change of temperature, is eliminated.

133. Phosphorus Is Undesirable in Boiler-plate Steel.—Although its presence makes a steel strong and hard and thus would seem desirable, these qualities are secured best through the medium of carbon. The reason is that phosphorus tends to make the material cold-short, that is brittle, when cold. Steel containing much phosphorus is particularly weak against shock and vibratory stresses. On this account, it may be considered the most harmful of the ingredients in steel boiler plate. It is for this reason that Bessemer process steels are undesirable for boilermaking. The method does not remove from the steel the phosphorus, which was originally in the pig iron.

134. Sulphur is detrimental to steel in various ways. Its principal effect is to impair the tenacity and ductility of the plate and to make it hot-short or brittle and difficult to work when hot.

135. Silicon in Mild Steel Makes It Harder.—There is but a small quantity present. Even this increases the hardness slightly, but without diminishing toughness or ductility and without affecting appreciably its tensile strength. This might, therefore, be regarded as a beneficial ingredient.

136. Manganese in Mild Steel Is a Hardening Agent.—Steel which contains a considerable proportion of this element acquires a peculiar brittleness and hardness which makes it difficult to cut with machine tools. Manganese has, however, a neutralizing effect on sulphur. It combines with sulphur

in the steel to form manganese sulphide. This component is less objectionable than the iron sulphide that would otherwise be formed. The presence of manganese might, therefore, be regarded as advantageous.

137. Chemical Properties for Steels Are Specified by the Boiler Codes.—The standard rules, those of the *American Society of Mechanical Engineers* for example, stipulate certain chemical properties for steels of various grades for plates, stays, rivets, and the like as shown in Table IV. Chemical properties and physical properties of the various grades of steel and iron are given.

138. Steel which is to be used in boiler manufacture should, steel castings excepted, be made by the open-hearth process. Castings may be electric furnace, open-hearth, crucible, or other process steel which affords a good product. Steel plates for any part of a boiler, when exposed to the fire or products of combustion and under pressure, should be of firebox quality. For any part under pressure, where firebox steel is not required, the plate may be firebox or flange quality. Manhole and handhole covers, other parts subjected to pressure, and braces and lugs may, when steel is used, be made of either firebox or flange quality. Steel bars used for braces should be of steel-bar stock. Stay bolts should be of either iron or steel stay-bolt stock.

139. The effect of high temperature on the strength of steel has been investigated by many experimenters within the past ten years. Much of this work has been reported in the transactions of the A.S.M.E. It has been found that high temperatures in additions to reducing the tensile strength also cause creep, *i.e.*, the steel tends to grow or stretch continuously under load. For temperatures above 700° the designer must check for temperature stress and permissible creep stress. (Creep stress is the stress at a given temperature which will result in stretch or elongation of the metal at a rate of 1 per cent of its length in 100,000 or 10,000 hr. of service.) Permissible stresses for temperatures above 700° are given in Table P-8 of the Boiler Code.

140. The physical tests for different classes of iron and steel comprise mainly, tests for (1) tenacity, (2) ductility,

TABLE IV.—PROPERTIES OF VARIOUS GRADES OF

Material	Where used	Chemical properties		
		Carbon, maximum per cent	Manganese, per cent	Phosphorus, maximum per cent
Steel Firebox (plate)	Parts exposed to fire and under pressure; also other plate	Less than ¾ in., 0.25; over ¾ in., 0.30	Less than ¾ in., 0.30–0.60; over ¾ in., 0.30–0.60	Acid, 0 05 Base, 0 035
Flange (plate)	For parts when fire-box quality is not specified; manhole and handhole covers		0 30–0 60	Acid, 0 05 Base, 0 04
Rivet	Rivets		0 30–0.50	0 04
Stay bolt	Stay bolts		0.30–0 50	0 04
Bars	Braces, and other bars not otherwise specified			Acid, 0 05 Base, 0 04
Tube, lap-welded and seamless boiler grade A	Lap-welded tubes and all seamless tubes	0 08–0 18	0 30–0 60	0 04
Castings Class A	Water leg and door-frame rings	0 45		Acid, 0 07 Base, 0 06
Class B	Cross pipes, headers, cross boxes, and pressure parts over 2-in. pipe sizes; mud drums; parts of superheaters; water leg and doorframe legs			0 05
Iron Rivet	Rivets			
Stay bolt	Stay bolts			
Refined wrought bars	Braces when welded			
Charcoal iron	Lap-welded tubes			
Malleable castings	Cross pipes, headers, cross boxes when pressure is less than 200 lb. per sq. in. and section within 7 by 7 in.			0 225
Gray castings Light	Boiler and superheater mountings, pipes, fittings, valves for temperature less than 450°F.; headers in water-tube boilers	(Any section less than ½ in. thick)		
Medium		(Castings not included in Light and Heavy)		
Heavy		(No section less than 2 in. thick)		

* From 1933 A.S.M.E. Boiler Code.

Steel and Iron Used in Boiler Construction*

	Physical properties					
Sulphur, maximum, per cent	Tensile strength, lb. per sq. in.	Yield point, lb. per sq. in.	Elongation, per cent, in 8 in. at least	Elongation, per cent, in 2 in.	Reduction area, minimum per cent	Special tests
0 04	55,000-65,000	0 5 tensile strength	$\frac{1,500,000}{\text{Tens. str.}}$ Min. = 20%			Bend homogeneity
0 05	55,000-65,000					Bend
0 045	45,000-55,000		$\frac{1,500,000}{\text{Tens. str.}}$			Bend, quench, and cold
0 045	60,000	26,000	Max. = 30%			Bend, quench, and cold
0.05	55,000-65,000	0 5 tensile strength	$\frac{1,500,000}{\text{Tens. str.}}$ Min. = 18%	26 min.		Bend
0 045						Flange flattening; hydrostatic
0 05		29,250		$\frac{1,600,000}{\text{Tens. str}}$ Min. 24%	$\frac{2,600,000}{\text{Tens. str.}}$ Min. 35%	Destruction
	48,000-52,000	0.6 tensile strength	28		45	Bend, cold, nick, and etch
	48,000-52,000		30		48	Bend, cold, quench, nick, and etch
	48,000-54,000		25		37	Bend, cold, hot, nick, and etch
						Flange; flattening; hydrostatic
0.06	50,000	32,500		10		
0 10	18,000					Transverse
0.10	21,000					
0.12	24,000					

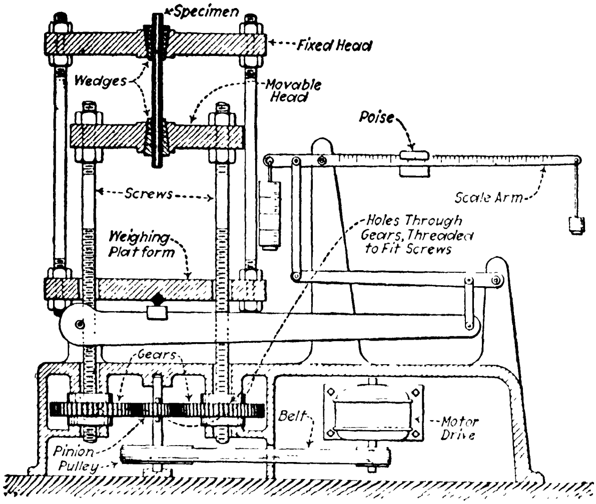


FIG. 97.—Diagram illustrating the principle of a standard tension and compression testing machine. (This is not intended to show actual construction.)

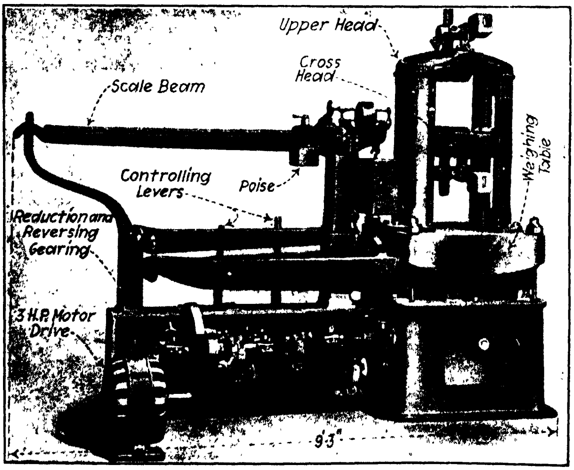


FIG. 98.—External appearance of a standard tension and compression testing machine (100,000 lb. or 50,000 kilogram capacity).

(3) elasticity, (4) malleability. Secondarily, they comprise tests for (5) homegeneity, (6) ability to resist compression.

141. The tension test consists in stretching the specimens in a testing machine (Figs. 97 and 98) until the specimen is pulled apart. The pull exerted, on the specimen, by the machine and indicated by it at the instant of rupture measures the ultimate tenacity, that is, the ultimate *tensile strength* of the material. During this test the elasticity and ductility of the steel may also be determined.

142. The method of conducting tension tests is as follows: The clamps (Fig. 99) of the extensometer are applied 9 in.

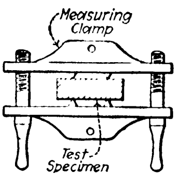


FIG. 99.—End view of plate (rectangular) test specimen showing measuring clamp applied.

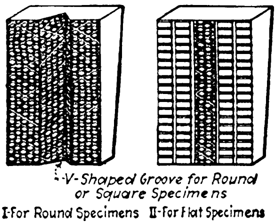


FIG. 100.—Forms of jaws or wedge grips for flat and round specimens.

apart on the specimen. The extensometer is an instrument for measuring elongation. It is made in numerous forms. The specimen with the extensometer on it is secured in the testing machine by clamping it in the wedge jaws (Figs. 100 and 101) with which every testing machine is equipped. Then the load is applied until the specimen is gripped. Now the distance between the index points of the extensometer clamps is measured with a micrometer (Fig. 102). Some extensometers have micrometers which form integral parts of the instrument. Extensometer measurements are observed periodically as the test proceeds.

143. In measuring the elongation (Fig. 102) the micrometer is placed against a point on the flange of the clamp on one end. Then the length of the micrometer is so adjusted that it will just touch the corresponding point on the other clamp.

As the load on the specimen is increased, the specimen stretches. The elongation (Fig. 103) is, within the elastic limit, proportional to the increase in load. The load which imposes the tension is continually increased by the motor or belt drive, which operates the testing machine, until the specimen finally breaks. The pounds load, imposed at any instant on the specimen, may be read from the scale arm in precisely the same

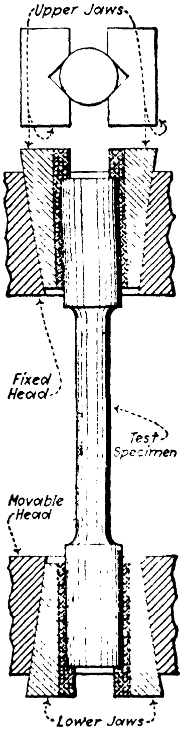


FIG. 101.—Showing arrangement of wedge shaped jaws of testing machine. (*Tinius Olsen Company.*)

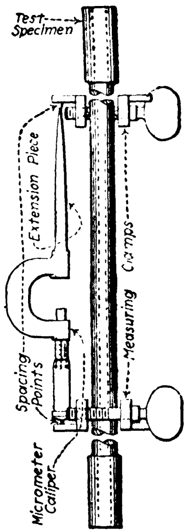


FIG. 102.—Edge view of round test specimen showing application of micrometer caliper.

way as the weight is ascertained from the beam of a platform scale in a grocery store. The "poise" is moved along the beam as the test progresses so that it always just counterbalances and indicates the load (tension) in pounds then imposed on the test specimen. Thus the load in pounds at which the specimen breaks may be determined.

144. The elastic limit of the material is attained during the progress of the test. This is the limit (the load, in pounds per square inch) beyond which the material cannot be stressed without producing a permanent change of shape. Precise determination of the elastic limit of a material demands extremely delicate adjustment and manipulation. Hence in specifying boiler steel, it is the practice to base requirements on the yield point (see definition below) rather than on the elastic limit. The yield point, which can readily be deter-

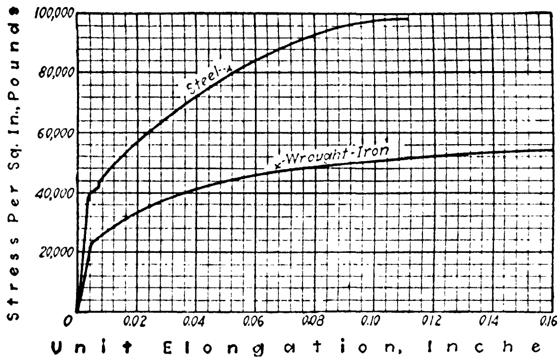


FIG. 103.—Typical stress diagram of tensile test of wrought iron and steel specimens.

mined, may be regarded as a sort of an approximate elastic limit. When a specimen is released from a tension less than its elastic limit, it will resume its original length.

145. The yield point is the stress under which the steel specimen begins to lengthen rapidly without a corresponding increase in the load. It is manifested by the scale poise on the testing machine indicating a diminished load (the testing machine beam drops) though the pull continues to be applied steadily. Thus the specimen under test begins to draw down rapidly somewhere near the middle of its length. Finally it breaks. The resulting fracture will have, usually, a cross-sectional area of about one-half to three-fourths of that which the original specimen had.

146. Jaws and wedges of a testing machine (Fig. 101) prevent stretching or flow of the material in the part of the speci-

men gripped by them. In fact the clamping effect of the jaws may hamper the elongation of the material for a distance of 2 or 3 in. beyond their edges. (The jaws of a machine having a capacity of from 50,000 to 100,000 lb. are 4 or 5 in. long.) The grip of the jaw (Fig. 100) will not be effective unless about 3 in. of the length of the standard specimen is seized at each end.

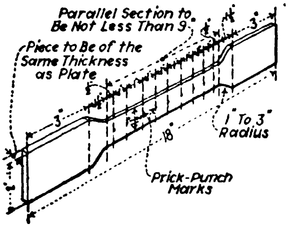


FIG. 104.—Steel-boiler-plate standard test specimen.

NOTE.—The test piece must have at least 8 in. of free length for measuring elongation. Also 2 or 3 in. must be allowed at each end for the flow of the material and from 3 to 5 in. must be provided for clamping in the jaws. Thus it follows that the specimen must have an over-all length of from 18 to 24 in. as shown in Fig. 104. The jaws have filelike flat faces (Fig. 100) for clamping flat specimens or V-notched faces for gripping round specimens (Fig. 105).

147. The tensile strength required for different grades of boiler steel as provided in the A.S.M.E. Boiler Code is given in preceding Table IV.

148. Tension-test specimens (Fig. 104) of boiler plate are made from strips cut from the plates (Fig. 106) as they come from the rolls. The specimen is usually cut out (Fig. 106 *I*)

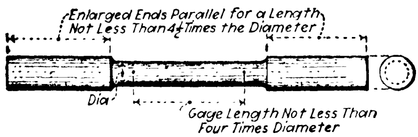


FIG. 105.—Standard test piece for round material.

in the direction of the longitudinal axis of the ingot from which the plate was rolled. But where the plate is to be so used that its transverse axis will lie in the circumference of the boiler shell, the test specimen is then cut out (Fig. 106–*II*) at right angles to the axis of the ingot from which the plate was rolled.

NOTE.—The edges of each test specimen are milled down to a uniform cross-sectional area of not less than ½ sq. in. for a space of about 9 in.

in the middle of the over-all length of the specimen, which is about 18 in. This space is marked off as delineated in Fig. 104.

149. The resilience of a material is its capacity to return the work which has been imparted to it. It is measured in foot-pounds. It is, practically speaking, equivalent to the potential energy stored up in the strained specimen. The amount of resilience is, practically, equal to the work—foot-pounds—required to deform the specimen, below the elastic limit, from zero stress to some specified stress.

150. The modulus of resilience or unit resilience is the elastic energy stored up in a cubic inch of strained material at the elastic limit. It is, essentially, the work done on a unit volume (1 cu. in.) in stressing it to the elastic limit. It is, therefore, equal to half the product of the elastic-limit strength multiplied by the corresponding unit deformation.

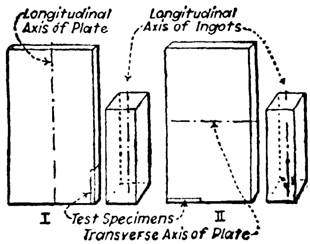


FIG. 106.—Showing location, in plate, of tension test specimens.

Example.—India rubber is very elastic because it will endure great deformation and yet return to its original form. But it is not very resilient because it has not, relatively, great capacity for resisting work. Steel is very resilient and quite elastic also. Steel will, within its elastic limit, endure deformation and yet return to its original form, hence it is elastic. It is very resilient because it requires much work to deform it, which work the steel—a spring for example—will give back if the force deforming it is released.

151. The ductility of the material is determined by the elongation of the specimen at the instant of failure. A material which can be stressed far beyond the elastic limit, undergoing during this stressing a considerable permanent deformation, is said to be "ductile." When the tensile stress exceeds the elastic limit the specimen becomes permanently lengthened. If it is of very ductile steel, it may stretch to 1.3 times its original length before breaking.

152. Compressive or crushing stresses in the section of metal between the edge of the sheet and the rivet holes must be considered in designing boiler seams. The molecules of

the metal oppose greater resistance to compression than to tension. Mild steel has a compressive strength of about 95,000 lb. per sq. in. The compression tests are effected with the same machine (Figs. 97 and 98) that is used for tension tests.

153. In making the compression test, the specimen is placed between the two plates or platforms of the machine, which are forced together by the turning of the powerful screws at each side which are driven by power. Figure 107 illustrates the arrangement. In Fig. 108 is shown the typical result of a compression test on a mild-steel specimen. The graph of Fig. 109 illustrates the typical performance of a specimen undergoing a compressive test.

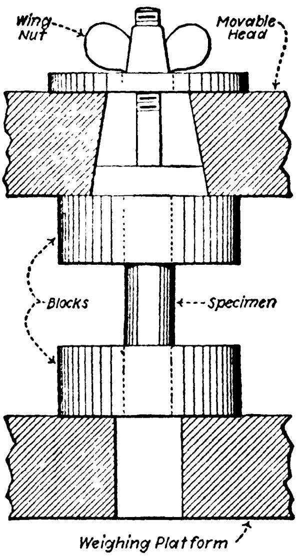


FIG. 107.—Arrangement of parts in compression test.

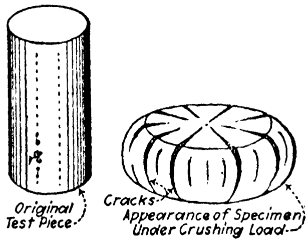


FIG. 108.—Steel specimen crushed in compression test

154. Malleability is a prime requisite of good rivet steel. The comparatively low percentage of carbon in steel of this class insures malleability. It also reduces the ability of the metal to resist shear and tension. Tension in the rivets of a boiler is, however, relatively unimportant. Furthermore, the cross-sectional area of rivets can, readily, be made sufficiently large to provide ample strength against shear.

155. The shearing strength of rivet steel is required (by the A.S.M.E. Code) to be at least 44,000 lb. per sq. in. in single shear (Fig. 110). In double shear or where cut through

simultaneously in two separate planes of cross section (Fig. 110), it must be at least 88,000 lb. per sq. in. Specimens for the shearing tests are made as illustrated in Fig. 110. The plates which hold the rivets to be tested, those in Fig. 110, should have an area of cross section sufficiently great that they will resist permanent distortion under any pulling stress less than the shearing strength of the rivet. The pounds tension under which the body of the rivet is cut or sheared through is its ultimate shearing strength. Rivet steel shall have a tensile strength of 45,000 to 55,000 lb. per sq. in.

156. The physical tests for boiler-plate steel are (1) tension (Table IV), (2) bend test, (3) homogeneity test applied only to firebox steel. Table III and the following sections recite the most important details concerning these tests and properties.

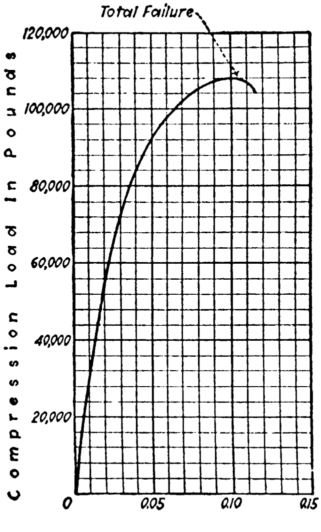


FIG. 109.—Typical stress curve of compression test.

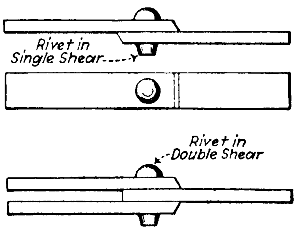


FIG. 110.—Specimens prepared for shearing tests.

157. Test Specimens Should Be Taken from the Finished Rolled Material (Fig. 106).—Tension-test specimens shall be taken longitudinally from the top and bottom corners of the finished rolled firebox material, and from the bottom only of flange material. Bend-test specimens shall be taken transversely from the middle of the top of the finished rolled material (Fig. 111). This latter requirement is based on the assumption that any possible segregation of the hardening

elements would tend to accumulate at the top of the ingot. Therefore, if a specimen taken from the location noted satisfies the prescribed test, it is reasonably certain that all other portions of the plate conform to the required standard.

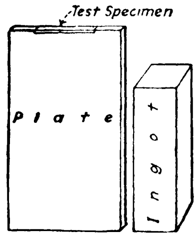


FIG. 111.—Showing location, in plate, of bend test specimen.

158. The bend test for boiler plate is as follows: For steel plate 1 in. or less in thickness, the specimen should bend cold (Fig. 112) 180 deg. without cracking around a pin of diameter equal to the thickness of the plate; for plate over 1 in. thick, the diameter of the pin should be twice the thickness of the plate.

159. The test for homogeneity of firebox steel should be made on a sample taken from a broken tension-test specimen. When the piece is nicked about 1/16 in. in three different places (see A.S.M.E. Code) and broken (Fig. 113) the fracture

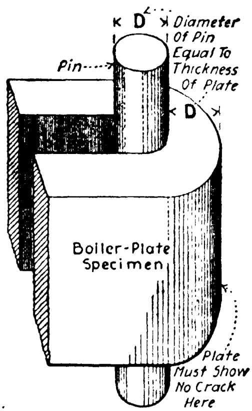


FIG. 112.—Bending test for materials thinner than one inch.

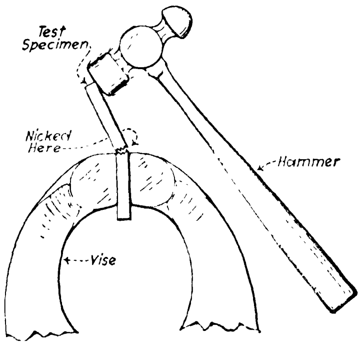


FIG. 113. How specimen is broken for homogeneity test.

should not show any single seam or cavity more than 1/4 in. long.

Small cavities are liable to be formed by gas bubbles in the molten ingot in the making of mild steel. Seams may also be found in the finished product. These may be due either to incomplete welding of

separate strata of the metal, or to slag penetrating between the strata. These defects are particularly objectionable in steel which is intended for shells, drums, and butt straps. The tensile stresses in these parts,

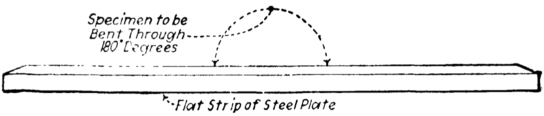


FIG. 114.—Test specimen before bending.

and in stayed flat sheets as well, demand that the steel be as nearly homogeneous as it is practicable to make it. A lesser degree of homogeneity may be permitted in rivet steel. Of recent years the processes of

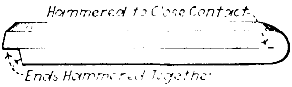


FIG. 115. Bending test applied to specimen of steel bar stock.

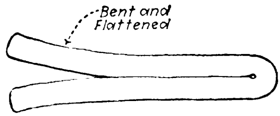


FIG. 116.—Bending tests on mild shaped specimens.

steel manufacture have been so perfected that little or no difficulty, due to these causes, is experienced.

160. Physical tests for rivet, stay bolt, and bar steels are (1) tension test (Table IV); (2) bend tests including (*a*) cold-

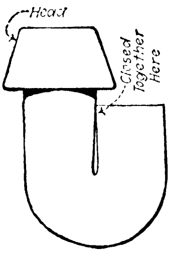


FIG. 117.—Rivet after being subjected to bending test.

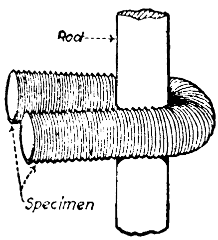


FIG. 118.—Obsolete test for malleability of stay-bolt stock.

bend test and (*b*) quench-bend tests; (3) a flattening test for rivet stock.

161. Specimens for tension and bend tests should, with the exception of those for steel-bar stock, be of full-size section of the stock. The specimen of steel bar, when the stock is less

than $1\frac{1}{2}$ in. thick, should be of the same thickness as the material after rolling. The piece may be machined to the form of Fig. 104 or with edges parallel. When the steel-bar stock is over $1\frac{1}{2}$ in. thick, the tension-test specimen may be of round section, similar to Fig. 105; the bend-test specimen may be machined down to 1 by $\frac{1}{2}$ in. section, and in testing bent around a pin 1 in. in diameter.

162. The cold-bend tests for rivet, stay bolt and bar steel are as follows: For stock 1 in. or less in thickness or in diameter, the cold specimen should bend through 180 deg. flat upon itself (Figs. 114, 115, 116, and 117) without cracking on the outside of the bent portion. For steel-bar stock over 1 in. and including $1\frac{1}{2}$ in. in thickness or diameter, the specimen should bend around a pin of a diameter equal to the diameter (Fig. 112) or thickness of the piece; when the material exceeds $1\frac{1}{2}$ in. in thickness or diameter, the pin should have a diameter of twice the thickness or diameter of the specimen.

NOTE.—Figure 118 shows a test sometimes made on stay-bolt material but the threading is not required by the A.S.M.E. Code. Figure 119 also shows a test for stay-bolt stock in tension which is not now generally specified. This is a test for toughness and tenacity. It is made by screwing each end of a sample bolt into nuts made from pieces of the plate to be stayed (Fig. 119) and imposing on the specimen so prepared a pulling load. If it fails by the bolt pulling apart, the load per square inch of net cross section at the instant of failure is taken as a measure of its strength. But if it fails owing to the nuts stripping the threads off the ends, the corresponding load is the measure of its strength. In the first case a tensile stress determines the result, in the second a shearing stress.

The product of one-half the thickness of the plate ($T \div 2$, Fig. 120) by the circumference C of the bolt at one-half the height of the thread gives the mean sectional area of the metal opposed to shear.

163. The Quench-bend Test Is Required for Rivet and Stay-bolt Steel.—The specimens are the same as those used for the cold-bend tests. After the specimens have been heated to a cherry red (in darkness), they are quenched in water which is at a temperature of from 80 to 90°F. They are then given the cold-bend test (Sec. 162). This test is intended to determine whether or not the material has hardening characteristics which would render its use dangerous in boiler construction.

164. A flattening test for rivet steel is also specified. The rivet head should flatten (Fig. 121) while hot to a diameter 2½ times the diameter of the shank, without cracking at the edges.

165. The tests for rivet and stay-bolt iron, and for refined wrought-iron bars are (1) tension tests (Table IV); (2) bend tests; (*a*) cold bend, (*b*) hot bend for rivet iron and refined

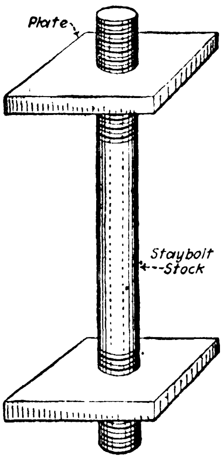


FIG. 119.—Obsolete method of preparing specimen of stay-bolt stock for tension test.

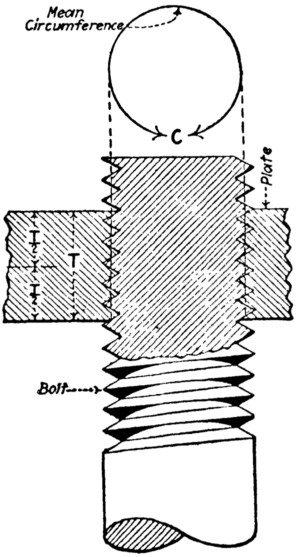


FIG. 120.—Computing area of metal opposed to shear.

wrought-iron bars, (*c*) quench bend for stay-bolt iron; (3) nick-bend tests; (4) etch tests.

166. The Tension and Bend for Iron Are Similar to Those for Steel.—It is impracticable to describe these tests in detail in the limited space available in this book. For further information the reader is referred to the A.S.M.E. Code.

167. A nick-bend test for iron is made by nicking the test piece with a chisel and then (Fig. 113) breaking the specimen.

This enables observation of the nature of the metal which should not be crystalline, with the exception that refined wrought-iron bars may show not more than 10 per cent of crystalline area.

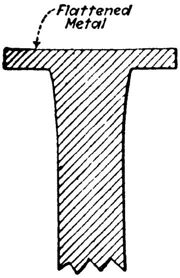


FIG. 121.—Rivet after flattening test.

168. The etch test on iron is made by polishing or grinding smooth the surface of a cross section of a specimen and then etching it with a solution composed of 10 per cent concentrated hydrochloric acid and 30 per cent concentrated sulphuric acid and 60 per cent water. This test is for determining if there is steel in the specimen; there should be none.

NOTE.—When etched as above, steel presents a uniform, glistening white surface which has a frosted white appearance. Wrought iron presents a black and gray striated appearance, the arrangement of which indicates the piling of the iron.

169. Tests for lap-welded and seamless boiler tubes are (1) flange test, (2) flattening test, (3) hydrostatic test, (4) tension test, (5) etch test.

170. The flange test for boiler tube is made to determine malleability.

Explanation.—For tubes not more than 6 in. in diameter, having a thickness less than 10 per cent of the outside diameter provided the thickness does not exceed No. 6 B.W.G. (0.203 in.), a test specimen not less than 4 in. in length (Fig. 122) shall have a flange turned over at right angles to the body of the tube without cracking or showing any flaw. The flange shall have a width, measured as in Fig. 122, of 0.1 the outside diameter of the tube for charcoal-iron tubes $3\frac{3}{4}$ to 5 in. in diameter and a width of $\frac{1}{2}$ in. for larger tubes. For open-hearth tubes and for smaller diameters see A.S.M.E. Boiler Code. For other tubes the flange test is not required. In making the flange test, the tube may first be flared with a flaring tool and then flattened down with a flatter.

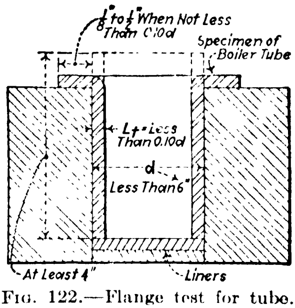


FIG. 122.—Flange test for tube.

171. A Flattening Test Is Also Required of Boiler Tubes.—A test specimen $2\frac{1}{2}$ in. long should stand flattening (Fig. 123)

between two parallel plates until the distance between the plates is not over five times the thickness of the tube wall, for a lap-welded tube; the weld should be at the point of maximum bend. For seamless tubes the distance between plates should be three times the wall thickness. Cracks or flaws should not appear.

NOTE.—In Fig. 124 are shown some specimens of lap-welded steel boiler tubes (National Tube Company) as they appeared after pressure had been applied to them endwise in a hydraulic press. The specimens withstood this special test without fracturing.

172. The hydrostatic test for boiler tubes is made by applying internal pressure. Tubes under 5 in. in diameter should

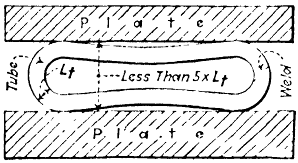


FIG. 123.—Flattening test for boiler tubes.

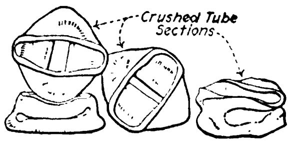


FIG. 124.—Tested sections of steel boiler-tubes.

stand 1,000 lb. per sq. in. Tubes over 5 in. in diameter should stand a pressure of 800 lb. per sq. in., providing that the fiber stress in the metal does not exceed 16,000 lb. per sq. in., in which case the testing pressure should be

$$P = \frac{32,000t}{D} \qquad \text{(pressure, lb. per sq. in.)} \qquad (1)$$

where P = test pressure, in pounds per square inch.
t = thickness of tube wall, in inches.
D = outside diameter of the tube, in inches.

Lap-welded tubes should be struck near both ends with a 2-lb. steel hammer while the pressure is being applied.

173. Etch test for charcoal-iron tubes is made as previously indicated for rivets and should not show the presence of steel. The weld should show very distinctly.

174. Cast steel is a grade of steel which can be cast into regular shapes by pouring the molten metal into molds. It contains (Table IV) about the same proportion of carbon as does mild steel. This is necessary to secure a condition of

fluidity that will insure a ready flow of metal into all parts of the mold. Cast steel was formerly used to some extent in sectional boilers as a material for headers. However, its application for this service has been practically discontinued. The difficulty of producing homogenous castings is the chief objection against the use of cast steel in boiler construction. When made without defects, however, the castings are very tough and tenacious.

175. Formerly Much Cast Iron Was Used in Boiler Construction.—Plain-cylinder boilers often had cast-iron heads. Sectional boilers also have been built of cast iron. Now, however, its utilization in boiler construction is restricted chiefly to mountings, supports, handhole and manhole plates, and for pressure-sustaining parts where the boiler is designed for comparatively low-pressure service, as for example, house heating. The best practice dictates the use of stamped or forged steel for irregularly shaped parts which are subjected to pressure.

176. Cast iron is a dangerous material to use when high steam pressures are concerned. This is chiefly because it is impossible to determine from its external appearance the internal condition of a casting. Large slag holes and cavities made by gas bubbles may be hidden under an apparently sound exterior surface. Also there may be severe local strains (on account of unequal cooling in the mold) in the grain of the metal. Even where castings are sound, there is still the danger of sudden cracking caused by the extreme variations of temperature prevailing in the generation of steam.

NOTE.—About the only characteristic of cast iron to recommend it as a boiler material is its practical imperviousness to ordinary corroding elements.

177. Malleable cast iron is produced by annealing ordinary cast-iron castings. This is done by raising the temperature of the castings to a red heat while in contact with oxide of iron which may be in the form of red hematite ore. Thereby a large proportion of the carbon in the castings is removed, by the process of chemical reduction. Thus malleability results from the expulsion of much of the original carbon content.

Malleable iron is allowed (A.S.M.E. Boiler Code) for use in cross pipes, headers, and cross boxes for pressures up to 350 lb. per sq. in., provided that the cross section of such parts will fall within a 7- by 7-in. rectangle.

NOTE.—The tensile strength of malleable iron is from 37,000 to 50,000 lb. per sq. in. It can, unless too thick, be bent and worked to some extent like wrought iron. The decarbonization seldom extends to a depth of more than ¼ in. from the outer surface of the casting. High temperatures impair its strength.

178. Tests on steel, gray-iron, and malleable castings (Table IV) cannot be discussed fully here. Steel castings are subjected to tension, bend, and destruction tests. Gray-iron castings are tested in tension and transversely (using an "arbitration bar"). Malleable castings are also tested in tension and transversely. (See the A.S.M.E. Boiler Code for details.)

179. Copper has characteristics which commend it as a boiler material for certain services. It is very ductile and malleable. It is a good conductor of heat and resists oxidation. The tensile strength of copper plates is about 34,000 lb. per sq. in. Copper is used largely in Europe for stay bolts and for the fireboxes of locomotive boilers. It has long since become obsolete in the United States for these purposes. Its high cost prohibits its use as a staple material for boilermaking. But even so, it is generally inferior to mild steel for the all-round purposes of boiler construction.

180. Bronze is an alloy of copper and tin. Its use in boiler work is confined to parts of valves and accessories where toughness and ability to resist corrosion are specially desirable qualities. The tin improves the fluidity of the molten alloy. It also increases the hardness of the finished casting, but diminishes its ductility.

181. Brass is an alloy of copper and zinc. It is widely used in the manufacture of boiler fittings. By reason of its comparative immunity from corrosion and incrustation, it is also regarded favorably as a material for the feed-water connections to steam boilers. In Europe it is used in the flue tubes of locomotive boilers. Its high heat conductivity recommends it for this service.

QUESTIONS ON DIVISION 8

1. Name some of the most common metals that are used in steam boiler construction.

2. Why is it important to consider carefully the quality of the materials that are used in steam boiler construction?

3. To what abuses may a boiler be subjected?

4. What kind of stresses are imposed upon the materials in a steam boiler?

5. What is wrought iron? Charcoal iron?

6. Describe briefly the manner of making a sheet of wrought iron.

7. What is low-carbon steel?

8. What process is used in making boiler steel?

9. What is done with steel when it comes from the furnace in a molten state?

10. State some distinguishing differences between wrought iron and steel.

11. What kind of steel is specified by the A.S.M.E. Boiler Code for boiler shells?

12. Name some of the principal elements found in mild steel Why are they in the steel?

13. What is meant by each of the following terms: *tenacity, elasticity, hardness, ductility, malleability, toughness, resilience, homogeneity?*

14. Describe the effect of carbon upon tensile strength of steel.

15. What makes boiler plate hard? Why not make boiler plate very hard?

16. What makes steel plate brittle?

17. What is the effect of the presence of phosphorus in steel?

18. What is the effect of sulphur in steel?

19. Discuss the effect of the presence of manganese in steel plate.

20. Mention some of the grades of steel used in making boilers and tell for what each grade is used.

21. What is the effect of high temperature upon steel?

22. Mention some of the important tests which are required for determining the fitness of iron and steel for use in a boiler.

23. How is a tension test made?

24. Describe the use of the extensometer.

25. What is meant by elastic limit? Yield point?

26. Which is the easier to determine and why, elastic limit or yield point?

27. Does the area of the cross section of a test piece change when it is pulled in two?

28. How are tension-test specimens obtained and prepared for testing?

29. What is meant by resilience?

30. How is ductility of a material determined?

31. How does the compressive strength of steel compare to its tensile strength?

32. What shearing strength is required of rivet steel?

33. What are the most important physical tests for boiler-plate steel?

34. How is a test specimen cut from plate steel and why?

35. How is the cold-bend test made on boiler-plate steel?

36. How may the homogeneity of a specimen be determined?

37. What tests are specified for rivet, stay bolt, and bar steels?

38. How should the specimen be prepared?

39. What differences are noted between cold-bend tests for steel plate and bar materials?

40. Describe a quench-bend test. Flattening test for rivet stock.

41. What is a nick-bend test? For what materials is it specified?

42. What is the purpose of an etch test?

43. How are boiler tubes tested?

44. How is cast steel used to make irregular forms for use in boiler construction? Why is it undesirable?

45. Why has cast iron been discarded as a material of which to make fittings for a boiler?

46. What good characteristic has cast iron?

47. How is malleable iron produced and what are its characteristics?

48. What tests are specified for steel castings? Gray-iron castings? Malleable castings?

49. Why is copper not used extensively in boiler construction?

50. What characteristics has bronze which makes it suitable for valve parts? Brass?

DIVISION 9

STRESSES IN AND STRENGTHS OF STEAM BOILERS

182. In considering the subject of stresses in and strengths of steam boilers, the principles involved will first be discussed as applying to an imaginary seamless cylindrical metal shell which is subjected to an internal pressure. Subsequently, it will be shown how these basic theoretical principles may be modified to permit their application to actual steam boiler design.

183. The technical meanings of the words "stress" and "strain" should be understood. The *stress* in a material is the internal resistance, offered by the molecules of the material, which opposes deformation or change of shape. Stress balances an external force which may be a push, pull, pressure, weight, or load. Force is the cause; stress is a result. Since the stress in a material is always just sufficient to counter balance the force which is applied, it is always equal, but opposite in direction, to the applied force. Stress is sometimes, though incorrectly, thought of as a load. But it will be understood here to be the internal resistance offered by the material to the load. Stress is measured in the same units as force, *i.e.*, in pounds or any other unit of weight. A *strain* is an alteration or deformation in size, form, or volume due to the application of a force. Strain may be considered as the stretch due to a load.

Example.—If a strip of boiler plate has a weight of 5,000 lb. suspended from one end, a stress of 5,000 lb. exists at every cross section of the strip.

Example.—If a steam pressure of 100 lb. per sq. in. be carried in a boiler, then there is a tendency to push the boiler apart, but the internal stress opposes the rupture of the shell. The shell is strained or stretched according to the intensity of the pressure in it.

184. Stresses in a boiler may be due to tension, compression, shear, flexure, torsion, or other combinations of tension,

compression, and shear may exist. A *tensile stress* is the stress opposed to pulling apart, or tension. A *compressive stress* is produced by a pushing together, crushing, or compression. A *shearing stress* is due to two forces tending to slide one part of a material past another. The forces act in parallel planes, usually very near each other. Such action is produced by shears which are used for cutting materials. *Flexural stress* is produced by bending. It is a combination of tensile and compressive stresses. It is the stress produced in a horizontal floor beam when it is loaded. The fibers above the neutral axis are in compression and those below the neutral axis are in tension. *Torsional stress* is produced by forces which cause a twisting or torque. Such a stress is a complex combination of tensile, compressive, and shearing stresses. Ordinarily, however, only the shearing stress is considered in torsion computations.

NOTE.—Since the pressure or force which is applied externally to a material is numerically equal to the stress which the pressure sets up in the material, it is often convenient to solve practical problems by considering the applied forces alone. This procedure will, in general, be followed in the succeeding discussion. But it should be remembered that the external force produces the internal stress and that it is the stress which is actually of importance when considering the strength of a material.

185. Steam Confined in Any Vessel Exerts the Same Pressure in Every Direction.—That is, the pounds push is the same against each and every square inch of the interior surface of the vessel. This is merely a statement of an observed fact.

186. All of the Stresses Produced by the Contained Steam Pressing on the Interior of a Shell May Be Resolved into Longitudinal and Transverse Stresses.—Thus while as above stated the steam pressure acts with equal force on every square inch of the interior surface of a shell (or any vessel), the combined effects of all of the resulting stresses may, if convenient (as will be explained), be considered as comprising only two resultant stresses: (1) a longitudinal stress, and (2) a transverse stress. The transverse stress is one in a direction at right angles to the axis of the shell. It resists forces (Fig. 125) which tend to tear the shell apart lengthwise. The longitu-

dinal stress is one in a direction along the axis of the shell. It resists forces (Fig. 126) which tend to tear the shell across or to separate it endwise.

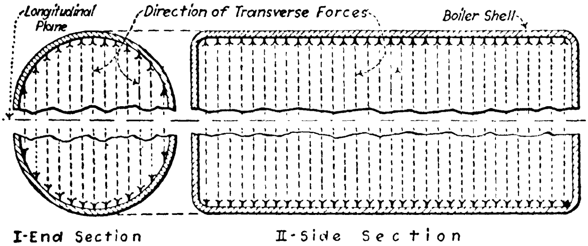


FIG. 125.—Illustrating "Transverse" forces. (The transverse forces produce transverse stresses.)

Thus any cylindrical vessel which is subjected to internal pressure may fail in either one of two ways: (1) Due to trans-

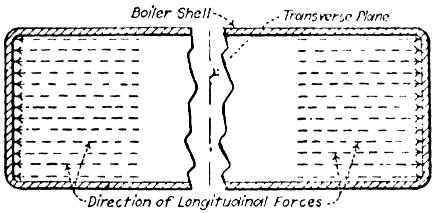


FIG. 126.—Illustrating "Longidutinal" forces. (The longitudinal forces produce longitudinal stresses.)

verse forces (Fig. 125) which tend to pull the shell apart sidewise or along a longitudinal plane; (2) Due to longitudinal

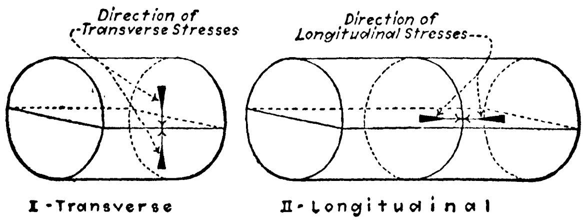


FIG. 127.—Showing how the transverse and longitudinal stresses are directed at right angles to each other.

forces (Fig. 126) which tend to pull the shell apart endwise or along a transverse plane. The transverse stress (which resists

longitudinal rupture) and its effects will be considered first, and then the longitudinal stress (which resists transverse rupture) will be discussed.

NOTE.—Since the transverse and longitudinal stresses act at right angles to each other (Fig. 127), they have no effect on each other. Therefore, they can and should be considered independently. In any cylindrical shell, the unit transverse stress is, as will be shown, always the

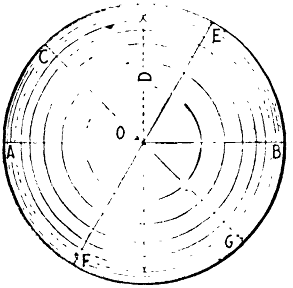


FIG. 128. Spherical shell.

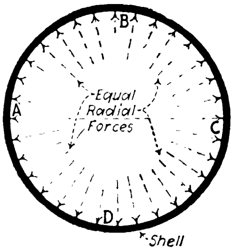


FIG. 129.—Illustrating direction of pressure on wall of cylindrical shell.

greater. In a spherical shell (Fig. 128) the "longitudinal" and the "transverse" stresses are, obviously, equal.

187. Steam in a cylindrical shell exerts a uniform pressure along radial lines (Fig. 129) against all points on the interior wall of the shell. (This is in addition to the longitudinal pressure which will be considered later.) The obvious action of the steam pressure, which thus acts radially from the axis of the shell and which is distributed uniformly over the interior surface as indicated by the arrows in Fig. 129, is to tend to preserve the cylinder from distortion from outside forces. It tends to round out any initial variations from the true circular contour. Hence the sheets of circular boiler shells and drums are self-supporting and require no bracing to resist distortion.

188. The transverse stress due to the steam in a cylindrical boiler tends to prevent rupture of its wall (Fig. 127–*I*) lengthwise, along any two diametrically opposite lines in its circumference. With a perfectly homogeneous seamless-metal vessel, a bursting pressure would tend to rupture the shell simultaneously along an infinite number of diametrically oppo-

site longitudinal lines. However, in an actual pressure vessel, there is always some line of least resistance to failure by lengthwise rupture. In a boiler shell, it would probably be along a seam.

189. It Is the Total Transverse Force, Acting in a Single Direction Only, That Is Effective in Rupturing the Shell Longitudinally.—Since the internal pressure acts radially in all directions (Fig. 129) from the axis of the cylinder, it might be assumed that the total force which tends to rupture the shell along AC would be the force against the upper half ABC plus the force against the lower half ADC. Such an assumption would be incorrect, as will be explained. The force which tends to rupture is that imposed on only one-half of the shell, either ABC or ADC. Note the following example and then the explanation below.

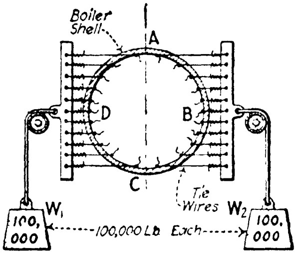


FIG. 130.—Transverse forces tending to pull shell asunder.

Example.—If the total transverse force imposed against ABC in Fig. 129 is 100,000 lb. and that against ADC is 100,000 lb., then the total pressure tending to split the shell along the line AC would be 100,000 lb. —not 200,000 lb.

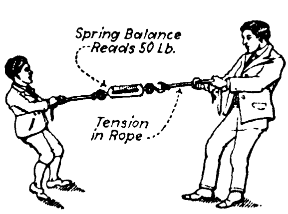


FIG. 131.—Every stress must be opposed by an equal and opposite force.

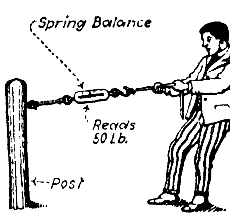


FIG. 132.—The post always pulls just exactly as much as does the man.

Explanation.—If the shell of Fig. 129 has an internal transverse force of 100,000 lb. imposed on each of its halves, it would be stressed the same as though two weights of 100,000 lb. each (Fig. 130) were pulling against one another through the shell. If the shell is of insufficient strength to withstand the force, it would pull apart on line AC. One

weight, W_1, constitutes merely the equal and opposite force with which every applied force must be resisted. Thus a man (Fig. 131) pulling against a boy who is holding the rope can pull no harder than the boy pulls against him. If the boy pulls 50 lb., the man must also pull 50 lb. A spring balance inserted in the rope would register 50 lb. If the man

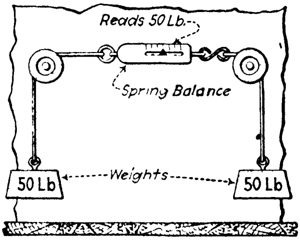


FIG. 133.—Equal and opposite forces.

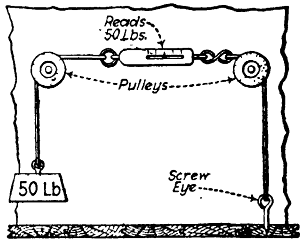


FIG. 134.—The screw eye always "pulls" just exactly as much as does the weight.

pulls harder than the boy, he will jerk the boy off his feet. In any case the stress, at any location in the rope, will equal only what one of them pulls. It will *not* be the sum of what both of them pull. The tension in the rope is 50 lb.—not 100 lb. A solidly set post might (Fig. 132) replace the boy. Then, when the man pulls with a force of 50 lb. against the post, the post would, obviously, be pulling against or holding him with a force of 50 lb. Again, a spring balance inserted in the rope would read 50 lb. Figures 133 and 134, wherein weights are substituted for the man and boy, respectively, and a screw eye in the floor for the post, further explain the situation.

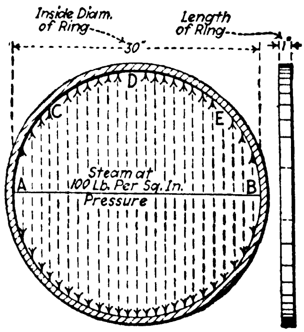


FIG. 135.—Illustrating direction of pressure acting to separate ring of shell plate on horizontal diameter.

190. The Area against Which the Pressure Is Assumed to Act, Is the Projected Area That Lies at Right-angles to the Direction of Pressure under Consideration.—Why this is true will be explained later. For the present, consider this example.

Example.—What is the total force tending to burst the ring in Fig. 134? This represents an imaginary 1-in. length of seamless steel shell of a boiler 30 in. in diameter under an internal pressure of 100 lb. per sq. in.

Solution.—Its projected area (see explanation below) is: 30 in. × 1 in. = 30 sq. in. The pressure on every square inch is 100 lb. Hence the force tending to burst the shell along the line AB is 30 × 100 = 3,000 lb. Obviously, since this force must be divided equally between the two sections of the shell in tension at A and at B, that at A is 1,500 lb. and that at B is 1,500 lb.

191. The projected area (Fig. 136) is in this connection the area crosswise to the direction of pressure. It is the area which the thing would present to the human eye if it were held up and looked at in the direction of the arrows (Fig. 135) which indicate the direction of the applied internal pressure. It is the area of the shadow (Fig. 136) which would be cast by rays

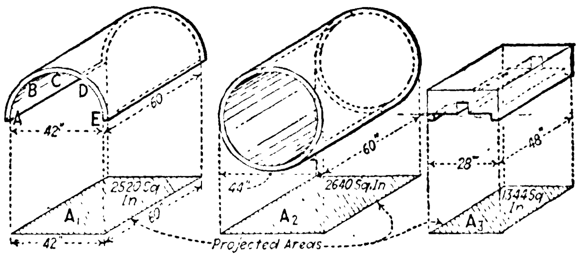


FIG. 136.—A_1, A_2, and A_3, are projected areas.

of light parallel to the direction of pressure. Thus, *the projected area, against which the pressure tending to burst a boiler longitudinally acts, is equal to the product of the internal diameter of the shell times the length of the shell under consideration.*

Example.—The projected area of the segment of the shell shown in Fig. 135 is 30 in. × 1 in. = 30 sq. in. The projected area of the segment of a half shell shown in Fig. 136–*I* is 42 in. × 60 in. = 2,520 sq. in. (Note the other examples in Fig. 136.)

192. Why the projected area is taken instead of the circumferential area against which the steam pressure acts will now be explained. At first glance, it might appear that the length of the half circle $ABCDE$ in Fig. 136–*I* should be multiplied by the length of the shell to obtain the area against which the effective pressure acts. It would be incorrect to do this.

Explanation.—Imagine that the inside surface of the ring of Fig. 135 is a series of infinitely small steps or corrugations. In Fig. 137, it is

apparent that it is only the pressure exerted against those surfaces of the serrations which lie parallel to the diameter AB that is effective in tending to burst the shell along the line AB. That is, it is only this "vertical component of the pressure in all directions" that tends to burst the shell

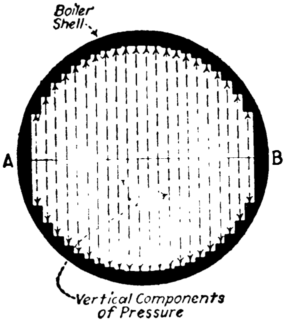


FIG. 137.—Illustrating vertical component of pressure on each semicircumference.

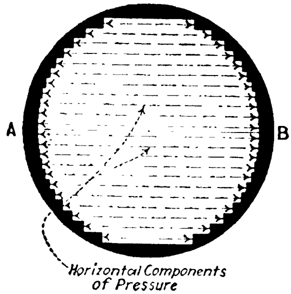


FIG. 138.—Illustrating horizontal component of pressure on each semicircumference.

along AB. The pressure acting on those surfaces of the corrugations which lie at right angles (Fig. 138) to the vertical diameter have no influence for or against separation along AB. This horizontal pressure is termed the *horizontal component of the total pressure*. Hence, as above explained, the area in Fig. 135 against which the bursting pressure is actually effective is 30 in. × 1 in. = 30 sq. in. In other words the only forces which tend directly to produce rupture are those which act at right angles to the line of rupture.

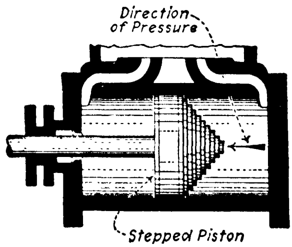


FIG. 139.—Stepped piston face.

Another viewpoint of this same situation (first suggested by Fred R. Low of *Power*) may be obtained by considering the pushing action of the steam against an engine piston. A piston, which has its outer face turned into steps (Fig. 139) would, obviously, have exactly the same area for the production of power as would a flat-surfaced one. Any steam thrust on one side, against a longitudinal face of one of the steps, would be counterbalanced by an exactly equal and opposite thrust from the other side. Only the steam thrust on the vertical faces is pushing the piston forward. Obviously, the total area of all of these vertical-face rings is precisely the same as that of a flat circle of the same over-all diameter. Some further illustrations of this same situation are given in Fig. 140.

The reason just outlined would be as true if the steps were a millionth or a hundred-millionth of an inch wide. It follows that the pushing pressure against a conical surface (Fig. 141) or a concave surface (Fig. 142) would be exactly the same as against a flat surface of the same over-all diameter. Hence it is apparent that it is the projected area in

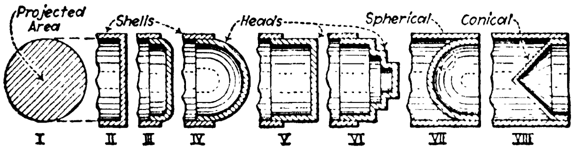


FIG. 140.—All have the same "projected area" as shown at *I*.

the direction of the pressure under consideration (not the actual superficial area) which is effective in producing a bursting stress in a boiler shell.

193. The formula for computing the total internal transverse rupturing force imposed on a cylindrical shell may now be developed. From the preceding statements it follows that to obtain this force it is merely necessary to multiply the

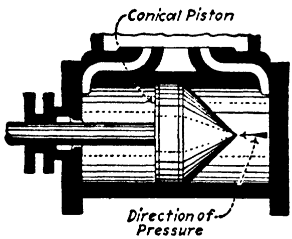


FIG. 141.—Conical piston face.

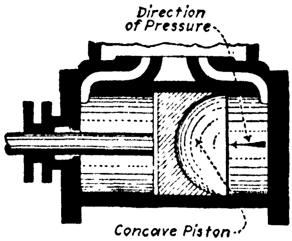


FIG. 142.—Concave piston face.

projected area in square inches by the pressure in pounds per square inch imposed on it. The projected area is equal to the product of the internal diameter and length. Expressing these operations as a formula:

$$P_T = dLP_{gt} \qquad \text{(force, lb.)} \qquad (2)$$

and

$$d = \frac{P_T}{LP_{gt}} \qquad \text{(internal diameter, in.)} \qquad (3)$$

and

$$L = \frac{P_T}{dP_{gt}} \qquad \text{(length, in.)} \qquad (4)$$

and

$$P_{gt} = \frac{P_T}{dL} \quad \text{(internal pressure, lb. per sq. in.)} \qquad (5)$$

where P_T = total force tending to rupture the shell lengthwise or longitudinally, in pounds.

d = internal diameter of the shell, in inches.

L = internal length of the shell in inches.

P_{gt} = internal pressure in pounds per square inch, gage.

Example.—Applying this formula to the example already given: What would be the total transverse force tending to rupture an imaginary boiler (Fig. 135) 1 in. long and 30 in. in internal diameter which is carrying steam at 100 lb. per sq. in. pressure? *Solution.*—Substitute in formula (2): $P_T = dLP_{gt} = 30 \times 1 \times 100 = 3{,}000$ lb.

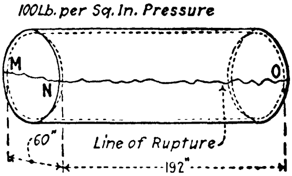


FIG. 143.—What transverse rupturing force imposed on shell 60 in. in internal diameter and 192 in. long inside with 100 lb. per sq. in. internal pressure.

Example.—The total force tending to rupture the shell of Fig. 143 along the plane *MNO* would be; $P_T = dLP_{gt} = 60 \times 192 \times 100 = 1{,}152{,}000$ lb.

194. The formula for computing the transverse stress set up in metal shell or along the line of a riveted seam will now be given. Obviously, resisting strength is offered by the two sections of metal (for example, *ABCD* and *EFGH* in Fig. 144) which extend longitudinally the length of the shell along the

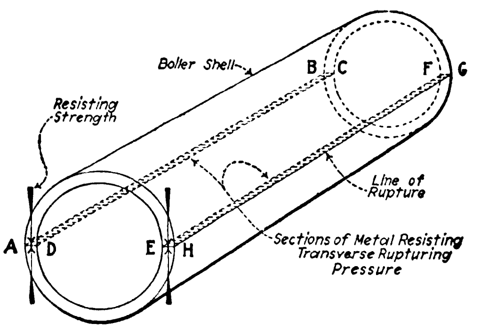


FIG. 144.—Sections of shell which resist bursting.

assumed plane of rupture. The stress (pounds per square inch) produced in the metal of these sections by the total force P_T tending to rupture the shell is equal to P_T divided by the area of the two sections of metal $ABCD$ and $EFGH$. Formula for the transverse stress due to pressure is

$$S = \frac{dP_{gt}}{2t} \quad \text{(lb. per sq. in.)} \tag{6}$$

where t = thickness of the metal in inches.

Derivation of the Above Formula.—Area of each section of metal is tL and the total area resisting P_T is $2tL$. Hence the stress $S = P_T/2tL$. Substituting for P_T its equivalent dLP_{gt} from equation (2), $S = dLP_{gt}/2tL$ The two L's cancel out, leaving:

$$S = \frac{dP_{gt}}{2t}$$

Example.—Taking values relating to Fig. 142 and assume metal thickness t = 0.25 in., the transverse stress will be $S = \frac{dP_{gt}}{2t} = \frac{60 \times 100}{2 \times 0.25} =$ 12,000 lb. per sq. in.

195. In boiler calculations no recognition is accorded to the resisting strength which is offered against transverse pressure by the ends or heads of the cylinder. It is apparent that the ends of the cylinder, particularly when they are flat, must increase to some extent the resisting strength against transverse pressure. The ends act like stays. But it is impossible to compute precisely the amount of the strength increase thus afforded. Hence in practice it is disregarded entirely.

NOTE.—It is often assumed in practice that the possible increase in strength, due to the ends, offsets indeterminate elements of weakness, such as poor joints, imperfect metal, corrosion, and improper staying.

196. The formula for computing the safe total resisting strength of a seamless cylindrical shell against transverse pressure (its derivation is given below) is:

$$S_{Tt} = \frac{2tU_tL}{f} \quad \text{(safe resisting strength, lb.)} \tag{7}$$

and

$$t = \frac{S_{Tt}f}{2U_tL} \quad \text{(plate thickness, in.)} \tag{8}$$

and

$$U_t = \frac{S_{Tt}f}{2tL} \quad \text{(tens. str., lb. per sq. in.)} \tag{9}$$

and

$$L = \frac{S_{Tt}f}{2tU_t} \quad \text{(length, in.)} \tag{10}$$

also

$$f = \frac{2tU_tL}{S_{Tt}} \quad \text{(factor of safety)} \tag{11}$$

where S_{Tt} = safe resisting strength of the shell against internal transverse bursting pressure, in pounds.

t = thickness of the plate, in inches.

U_t = ultimate unit tensile strength of the metal, in pounds per square inch.

L = internal length of the shell, in inches.

f = assumed factor of safety.

The derivation of the above formulas is this: First, note (Sec. 195) that the resisting strength of the ends is disregarded. Each square inch of the shell-plate section which is in tension offers a certain number of pounds resistance to rupture. Hence to obtain the total *ultimate* resistance to bursting, it is merely necessary to compute the total number of square inches of metal which opposes the bursting pressure, and multiply this area by the number of pounds per square inch which the metal will sustain. Then to obtain the safe resisting strength against bursting, the ultimate value, computed as above described, must be divided by the factor of safety (see Sec. 197 below). Expressing all of these operations in one equation, the working formula results, thus:

$$S_{Tt} = \frac{2tU_tL}{f} \quad \text{(resisting strength, lb.)} \tag{12}$$

Example.—If the plate metal in the boiler of Fig. 143 is ½ in. thick and has an ultimate unit tensile strength of 50,000 lb. per sq. in., what, assuming a factor of safety of 5, is the total transverse bursting force which the boiler will safely sustain? *Solution.*—The length is 192 in. Substitute in formula (7) above: $S_{Tt} = 2tU_tL/f = (2 \times 0.5 \times 50{,}000 \times 192) \div 5 = 1{,}920{,}000$ lb.

197. A "Factor of Safety" Is the Number Which Expresses the Ratio Which the Ultimate Strength of the Material Bears to the Safe Permissible Stress Set up in the Material.

Example.—If the ultimate tensile strength of steel boiler plate is 55,000 lb. per sq. in., and a boiler is so designed that only 11,000 lb. per sq. in. stress is set up in that plate, then: 55,000 ÷ 11,000 = 5 = factor of safety. Again, if the ultimate strength of the material is 55,000 lb. and the existing stress is 55,000 lb., then 55,000 ÷ 55,000 = 1 = factor of safety. That is, a factor of safety of 1 means that the material under consideration is stressed to its ultimate tensile strength. In effect, then, a factor of safety of 1 is, in reality, no factor of safety at all.

198. Factors of safety for steam boilers as recommended by the A.S.M.E. 1933 Boiler Code are as follows:

	Factor of Safety
Boilers in service when code becomes effective: after first year following code adoption and during ensuing four years	4.0
Boilers in service when code becomes effective: after fifth year following code adoption	4.5
Power plant, new installations	5.0
Secondhand boilers	5.5

199. The formulas for computing the safe-unit transverse pressure which may be imposed on a seamless, cylindrical shell, or the safe-plate thickness and diameter, or the safety factor may, as explained below, be derived from the preceding equation. The working formulas are:

$$P_{ot} = \frac{2tU_t}{df} \quad \text{(safe internal pressure, lb. per sq. in.)} \qquad (13)$$

hence

$$t = \frac{dfP_{ot}}{2U_t} \quad \text{(plate thickness, in.)} \qquad (14)$$

also

$$U_t = \frac{dfP_{ot}}{2t} \quad \text{(ultimate tens. str., lb. per sq. in.)} \qquad (15)$$

furthermore

$$d = \frac{2tU_t}{P_{ot}f} \quad \text{(diameter, in.)} \qquad (16)$$

and

$$f = \frac{2tU_t}{P_{ot}d} \quad \text{(factor of safety)} \qquad (17)$$

where P_{gt} = safe internal working pressure, in pounds per square inch.

t = safe thickness of shell, in inches.

U_t = ultimate unit tensile strength of the shell material, in pounds per square inch.

d = safe internal diameter of the shell, in inches.

f = factor of safety.

The derivation of the above formulas is this: If a cylindrical shell is stressed transversely by internal pressure just to the safe resisting strength of its material, then under this condition, the ***total internal transverse rupturing force*** must just equal the safe resisting strength. That is, using the symbols specified above:

$$P_T = S_{Tt} \qquad \text{(lb.)} \tag{18}$$

Now substitute for P_T from equation (2) its equivalent, dLP_{gt}, and for S_{Tt} its equivalent, $2tU_tL/f$ from (7). Then:

$$dLP_{gt} = \frac{2tU_tL}{f} \qquad \text{(2) and (7) in (18)} \tag{19}$$

$$P_{gt} = \frac{2tU_tL}{dLf} \qquad \text{(lb. per sq. in.)} \tag{20}$$

The two L's cancel out, leaving:

$$P_{gt} = \frac{2tU_t}{df} \qquad \text{(safe internal pressure, lb. per sq. in.)} \tag{21}$$

Example.—What would be the safe working pressure against longitudinal rupture for a seamless boiler, 60 in. in diameter, made of ½-in. steel plate, having a tensile strength of 50,000 lb. per sq. in.? Assume a safety factor of 5. *Solution.*—A substitute in formula (13): $P_{gt} = (2 \times t \times U_t) \div (d \times f) = (2 \times 0.5 \times 50{,}000) \div (60 \times 5) = 166.7$ lb. per sq. in. = working pressure.

200. The method of computing the bursting internal pressure with any formula which contains a factor of safety, f, is merely to use a value of unity, or 1, as a factor-of safety value. This, obviously, is equivalent to allowing no factor of safety at all. When this is done the significance of the symbols in the formula is changed somewhat, as is explained in the following section.

201. The transverse bursting pressure of a seamless shell may be computed by using formula (13) with a factor of safety of unity, or 1. It then becomes:

$$P_{bt} = \frac{2tU_t}{d} \quad \text{(bursting pressure, lb. per sq. in.)} \quad (22)$$

where P_{bt} = transverse internal bursting pressure, in pounds per square inch.

All the other symbols have the same meanings as above specified except that they indicate dimensions and properties of the metal when it is stressed to the bursting point.

202. The above formulas do not hold for thick-walled cylindrical shells, like that of Fig. 145 for example. With the thick-walled shell, additional modifying factors are introduced which it is unnecessary to consider here. The above

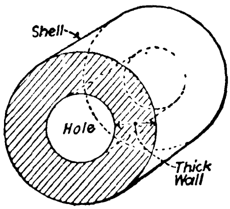


FIG. 145.—Thick-walled cylindrical shell.

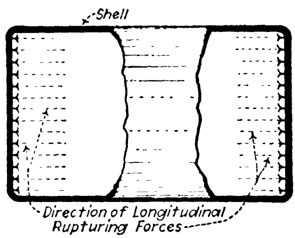


FIG. 146.—Illustrating direction of pressure acting to separate wall of cylindrical shell roundabout.

formulas will give correct results for shell thicknesses which are used in steam-boiler construction, excepting large boilers designed for very high pressure.

203. The longitudinal stress which is developed by the internal pressure (Fig. 146) will now be considered. An internal pressure not only (Sec. 186) tends, owing to its crosswise component, to split the shell lengthwise (Fig. 125), but it also tends, owing to its lengthwise component, to blow the heads off or rupture it roundabout, as indicated in Figs. 126 and 146. (If the section of metal which is stressed is insufficient, the shell will then be thus ruptured.) This lengthwise component produces in the metal of the shell the longitudinal stress which is now to be examined. The longitudinal internal-pressure component—due in a boiler to the contained steam—is, obviously, in a direction parallel to the axis of the cylinder.

NOTE.—These longitudinal forces act (Fig. 147) like two weights, one drawing on each head and tending to tear the heads from the shell. The effect of the longitudinal force in this respect is similar to that of

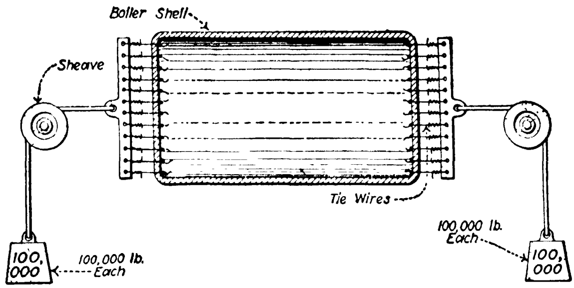


FIG. 147.—Longitudinal forces tending to pull the shell asunder.

the transverse force illustrated in Fig. 130 and described in accompanying paragraphs, which should be reviewed.

204. The formula for computing the total internal longitudinal rupturing force imposed on a cylindrical shell may now be derived. It is apparent from what has preceded that if the projected area (Fig. 140) of the head of the shell in square inches—which is the same as the internal cross-sectional area of the shell—be multiplied by the im-

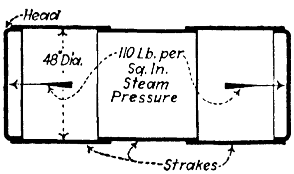


FIG. 148.—What force against heads? (NOTE.—A "Strake" is a shell plate.)

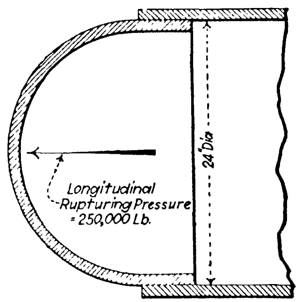


FIG. 149.—What internal pressure?

posed internal pressure in pounds per square inch, the total longitudinal rupturing force will be the result. Now the area of any circle equals its *diameter squared times the constant* 0.7854. Hence expressing the entire operation in a formula:

$$P_L = 0.7854d^2P_{gi} \qquad \text{(total longitudinal force, lb.)} \qquad (23)$$

hence

$$d = \sqrt{\frac{P_L}{0.7854 P_{gl}}} \quad \text{(internal diameter, in.)} \qquad (24)$$

and

$$P_{gl} = \frac{P_L}{0.7854 d^2} \quad \text{(pressure, lb. per sq. in.)} \qquad (25)$$

where P_L = total longitudinal force, in pounds tending to blow the head from the shell.
d = internal diameter of the shell, in inches.
P_{gl} = internal pressure imposed on the shell, in pounds per square inch.

Example.—The force tending to blow the heads from the shell of Fig. 148 would be, using formula (23): $P_L = 0.7854 d^2 P_{gl} = 0.7854 \times 48 \times 48 \times 110 = 199{,}000$ lb. = total longitudinal rupturing force.

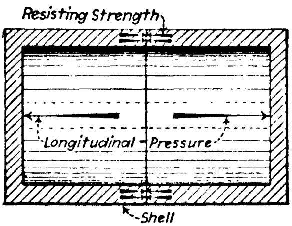


FIG. 150.—Illustrating resisting strength opposing longitudinal pressure.

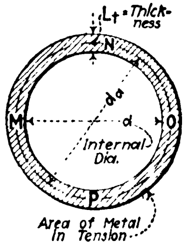


FIG. 151.—Area of metal in tension which resists longitudinal pressure.

Example.—The shell of Fig. 149 has an internal diameter of 24 in. The total force due to an internal pressure against the head tending to blow it out, is 250,000 lb. What is the internal pressure? *Solution.* Substitute in formula (25): $P_{gl} = P_L / 0.7854 d^2 = 250{,}000 \div (0.7854 \times 24 \times 24) = 552.6$ lb. per sq. in.

205. The resisting strength which a cylindrical steel shell offers to longitudinal rupturing pressure, such as that which occurs in a steam boiler owing to the confined steam, will now be considered. This tensile strength in a seamless shell would reside in the ring-shaped section of the metal (Figs. 150 and 151) extending circumferentially completely around the shell.

In an actual boiler, the riveted joint between the head and the shell would afford this resisting strength.

206. The formula for computing the safe total-resisting strength of a seamless cylindrical shell against longitudinal pressures follows. Its derivation is given below.

$$S_{Tl} = \frac{3.1416dtU_t}{f} \quad \text{(safe resisting strength, lb.)} \qquad (26)$$

and

$$d = \frac{S_{Tl}f}{3.1416tU_t} \quad \text{(internal diameter, in.)} \qquad (27)$$

and

$$t = \frac{fS_{Tl}}{3.1416dU_t} \quad \text{(thickness, in.)} \qquad (28)$$

also

$$U_t = \frac{fS_{Tl}}{3.1416dt} \quad \text{(ultimate tens. str., lb. per sq. in.)} \qquad (29)$$

hence

$$f = \frac{3.1416dtU_t}{S_{Tl}} \quad \text{(safety factor)} \qquad (30)$$

where S_{Tl} = safe resisting strength against longitudinal pressure, in pounds.
d = safe inside diameter of the shell, in inches.
U_t = ultimate unit tensile strength of the metal of the shell, in pounds per square inch.
t = safe thickness of the shell, in inches.
f = assumed factor of safety.

The derivation of the above equations is this: Each square inch of the annular section *MNOP* (Fig. 151) of the shell is capable of exerting a resistance against rupture equal to the ultimate tensile strength in pounds per square inch of the metal composing the shell. Therefore, if the area of this annular section, expressed in square inches, be multiplied by the unit ultimate tensile strength of the metal, the ultimate resisting strength will be the result. Then if the ultimate strength thus found be divided by the assumed factor of safety, the *safe* resisting strength will be obtained. Now the area of any annular or ring-shaped section (Fig. 151) is its *average circumference times its thickness t.* To obtain average circumference, the average diameter d_a is multiplied by 3.1416. It is evident

from Fig. 151 that average diameter $d_a = (d + \frac{1}{2}t + \frac{1}{2}t) = (d + t)$. Hence area $= 3.1416(d + t)t$. Dividing by the assumed factor of safety f gives the safe load. Now expressing the complete operation as a formula:

$$S_{Tl} = 3.1416(d + t) \times \frac{tU_t}{f} \quad \text{(longitudinal resisting strength, lb.)} \quad (31)$$

But in practice, the thickness t added to the diameter, d results in a very small and inconsequential increase. Hence it is omitted and the working formula then becomes:

$$S_{Tl} = \frac{3.1416dtU_t}{f} \quad \text{(longitudinal resisting strength, lb.)} \quad (32)$$

Example.—The resisting strength of the steel shell of Fig. 152 against longitudinal pressure, assuming a factor of safety of 5 and a unit ultimate tensile strength of 50,000 lb. per sq. in. for the metal, would be: $S_{Tl} = (3.1416dtU_t) \div f = (3.14 \times 30 \times 0.5 \times 50{,}000) \div 5 = 471{,}000$ lb. (In practice, the constant 3.1416 is abbreviated to 3.14.)

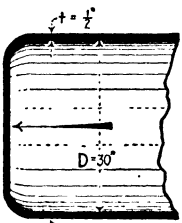


FIG. 152.—What is the longitudinal resisting strength of the shell?

207. The formulas for computing the safe longitudinal pressure which may be imposed on a seamless cylindrical shell, the safe plate thickness, the diameter, and the safety factor may, as explained below, be derived from the preceding equations, thus:

$$P_{gl} = \frac{4tU_t}{df} \quad \text{(safe internal pressure, lb. per sq. in.)} \quad (33)$$

hence

$$t = \frac{dP_{gl}f}{4U_t} \quad \text{(safe thickness, in.)} \quad (34)$$

and

$$U_t = \frac{dP_{gl}f}{4t} \quad \text{(ultimate tens. str., lb. per sq. in.)} \quad (35)$$

$$d = \frac{4tU_t}{P_{gl}f} \quad \text{(diameter, in.)} \quad (36)$$

where P_{gl} = safe longitudinal internal pressure, in pounds per square inch.

t = thickness of shell material, in inches.

U_t = ultimate unit tensile strength of the material, in pounds per square inch.
d = internal diameter of shell, in inches.
f = factor of safety.

The derivation of the above equations is this: If a cylindrical shell is stressed longitudinally (Fig. 150) by internal pressure just to the safe resisting strength of the material, then under this condition the *total internal longitudinal rupturing force* must be just equal to the *safe resisting strength against longitudinal pressure.* That is, using the symbols specified above, $P_L = S_{Tl}$. Now for P_L substitute from (23) its equivalent, $0.7854d^2P_{gl}$, and for S_{Tl} its equivalent, $3.1416dtU_t/f$ from (26). Then:

$$0.7854d^2P_{gl} = \frac{3.1416dtU_t}{f} \qquad \text{(23) and (26) in (37)} \qquad (37)$$

Then solving for P_{gl}:

$$P_{gl} = \frac{3.1416dtU_t}{0.7854fd^2} \qquad \text{(lb. per sq. in.)} \qquad (38)$$

This simplifying the above to the working formula:

$$P_{gl} = \frac{4tU_t}{df} \qquad \text{(safe pressure, lb. per sq. in.)} \qquad (39)$$

Example.—What is the safe steam pressure against roundabout rupture that may be carried in a seamless steel shell 60 in. in diameter made of ½-in. thick steel plate which has a tensile strength of 50,000 lb. per sq. in.? Assume a safety factor of 5. *Solution.*—Substitute in formula (33): $P_{gl} = 4tU_t/df = (4 \times 0.5 \times 50{,}000) \div (60 \times 5) = 333.3$ lb. per sq. in.

208. The unit longitudinal bursting pressure of a seamless shell may be computed with formula (33) by assuming a factor of safety of 1. The working formula then becomes:

$$P_{bl} = \frac{4tU_t}{d} \qquad \text{(lb. per sq. in.)} \qquad (40)$$

where P_{bl} = internal pressure, in pounds per square inch, which would cause bursting of the shell in a roundabout direction. All the other symbols have the same meanings as above specified, except that they indicate dimensions and properties of the metal when stressed to the bursting point.

209. A Seamless Homogeneous Cylindrical Shell of Uniform Thickness Is Twice as Strong against Roundabout Rupture as

against Longitudinal Rupture.—That is, the unit stress on the circumferential seams is half that on the longitudinal seams. The formula (40) for longitudinal internal bursting pressure which causes roundabout rupture is: $P_{bl} = 4tU_t/d$. The transverse internal bursting pressure which causes longitudinal rupture is [formula (22)]: $P_{bt} = 2tU_t/d$. Obviously, the bursting pressure in the first case is just twice as great as that in the second. That is, it requires twice the pounds-per-square-inch internal pressure to rupture the shell roundabout as it does to rupture it lengthways. But in any given shell the same pound-per-square-inch steam pressure acts

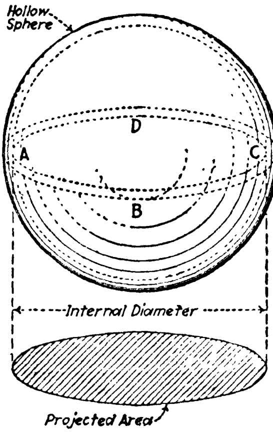


FIG. 153.—Hollow sphere.

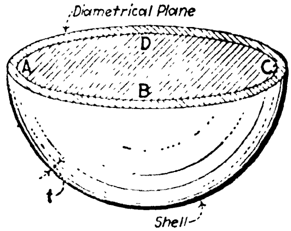


FIG. 154.—Illustrating diametral plane.

both longitudinally and transversely. Hence there is in any given cylindrical shell twice as much load imposed on the longitudinal as on the transverse seams. This is the reason for making the circular seams of boilers single-riveted, when the longitudinal seams are double-riveted and even triple-riveted. It is merely a matter of preserving the proportions.

Example.—The same dimensions are used in the examples under formulas (13) and (33). For the safe transverse pressure, using formula (13), the result is 166.7 lb. per sq. in. For the safe longitudinal pressure, using formula (33), the result is 333.3 lb. per sq. in. The first value is just half the second. Other similar examples will further verify the rule.

210. The stresses existing in a spherical shell due to an internal pressure and the resisting strength which the shell offers thereto may be computed readily. Obviously, from

the preceding discussion, the internal stress due to internal pressure in a hollow sphere (Fig. 128) is the same on any diametral plane, AB, CD, or EF. The formulas specified for the computation of longitudinal pressures in a cylindrical shell and of the resistance against these pressures apply, without modification, to the hollow sphere. The projected area (Fig. 153) is the internal area on any diametral plane, as for example, $ABCD$, Fig. 154.

Example.—What safe internal steam pressure will the steel hollow sphere of Fig. 155 safely sustain? Its internal diameter is 24 in. Thickness of plate, which has an ultimate tensile strength of 50,000 lb. per sq. in., is ½ in. Assume factor of safety of 5. *Solution.*—Substitute in formula (33): $P_{gl} = 4tU_t/df = (4 \times 0.5 \times 50{,}000) \div (24 \times 5) = 833$ lb. per sq. in.

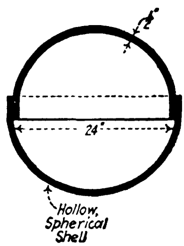


FIG. 155.—What pressure will the shell sustain?

211. How the foregoing theoretical principles relating to seamless cylindrical shells are applied in the design of actual power-plant steam boilers will now be explained. The outstanding difference between the previously discussed ideal seamless cylindrical shells and an actual boiler is that in the real boiler there must be riveted joints. It is impracticable to construct, without joints, a steam power-plant boiler. No riveted or welded joint can be as strong as the plates it joins, as will be explained in the next division. But, as also will be described, the percentage tensile strength, which a riveted joint has as compared with the tensile strength of the perfect plates which form the joint, can be determined readily. This percentage-strength value is called the *efficiency* of the joint. (Riveted-joint efficiencies are examined in detail in the succeeding division.) "Efficiency" in this connection may relate to the relative strength of some weak part of the boiler structure other than the joint. This situation is covered in detail in the A.S.M.E. Boiler Code.

212. To calculate the safe working steam pressure for a steam boiler of given dimensions, it is merely necessary to find by using a preceding formula the safe working pressure for a cylindrical shell of the same dimensions and made of the

same material as the boiler. Then this safe pressure for the seamless shell is multiplied by the efficiency of the weakest joint or other element to obtain the maximum allowable working pressure for the boiler. All of these operations are combined in formula (41) for computing the maximum allowable working pressure.

NOTE.—As explained above, the unit transverse stress in a cylindrical shell is always twice as great as the unit longitudinal stress. Hence the longitudinal riveted joint is probably the weakest member. Therefore, the efficiency of the longitudinal joint is, as specified below, employed ordinarily in the "maximum working pressure" calculation. But if the ligaments between the tube holes (par. 192 A.S.M.E. Boiler Code, 1933), are the weaker, their efficiency is used instead.

213. The A.S.M.E. Boiler Code formula for computing unit maximum allowable working pressures for power boilers, using the same symbols as those in the A.S.M.E. Code, is quoted below.

$$P_{MAW} = \frac{TS \times t \times E}{R \times FS} \quad \text{(pressure, lb. per sq. in.)} \tag{41}$$

hence

$$TS = \frac{P_{MAW} \times R \times FS}{t \times E} \quad \text{(tens. stress, lb. per sq. in.)} \tag{42}$$

and

$$t = \frac{P_{MAW} \times R \times FS}{TS \times E} \quad \text{(thickness, in.)} \tag{43}$$

and

$$E = \frac{P_{MAW} \times R \times FS}{TS \times t} \quad \text{(efficiency)} \tag{44}$$

and

$$R = \frac{TS \times t \times E}{P_{MAW} \times FS} \quad \text{(radius, in.)} \tag{45}$$

and

$$FS = \frac{TS \times t \times E}{P_{MAW} \times R} \quad \text{(safety factor)} \tag{46}$$

where TS = ultimate tensile strength of boiler plate stamped on shell plates, as provided for in the specifica-

tions of the material, in pounds per square inch.

t = minimum thickness of shell plates in weakest course, in inches.

E = efficiency, of longitudinal joint or of ligaments between tube holes (whichever is the lesser), expressed decimally.

R = inside radius of the weakest course of the shell or drum, in inches.

FS = factor of safety, or ratio of the ultimate strength of the material to the allowable stress, see Sec. 198 for factors of safety.

P_{MAW} = maximum allowable working pressure, in pounds per square inch.

Example.—What is the safe working steam pressure for a boiler 44 in. internal diameter having shell plates $^5/_{16}$ (0.3125) in. thick with an ultimate tensile strength of 55,000 lb. per sq. in. The efficiency of the weakest joint is 0.82, and a factor of safety of 5 is to be allowed. *Solution.*—Substitute in formula (41): $-P_{MAW} = (TS \times t \times E) \div (R \times FS) = (55{,}000 \times 0.313 \times 0.82) \div (22 \times 5) = 128$ lb. per sq. in. = maximum allowable working pressure.

Example.—What is the largest safe internal diameter which a boiler to carry 150 lb. per sq. in. steam pressure can have, if it is made of $^3/_8$-in. plate having an ultimate tensile strength of 50,000 lb. per sq. in.? The weakest joint is the longitudinal quadruple-riveted joint which has an efficiency of 0.937. Factor of safety is to be 5. *Solution.*—Substitute in formula (45): $(R = TS \times t \times E) \div (P_{MAW} \times FS) = (50{,}000 \times 0.375 \times 0.937) \div (150 \times 5) = 23.4$ in. = radius. Hence the diameter would be: $2 \times 23.4 = 46.8$ in. or practically 47 in.

QUESTIONS ON DIVISION 9

1. What is the technical meaning of the word *stress? Strain?*

2. What kinds of stresses are found in a boiler?

3. If the pressure exerted by the steam in a boiler is such that it presses with a force of 35 lb. on a square inch of the end of the boiler, what will be the pressure exerted upon a square inch of the side of the boiler?

4. What are transverse and longitudinal stresses in a boiler shell?

5. How may a cylindrical shell fail when the pressure inside becomes great?

6. Why are the cylindrical plates of a cylindrical boiler shell self-supporting?

7. How does transverse pressure due to steam in a cylindrical vessel tend to rupture it?

8. Explain the fact that the pressure which tends to rupture a shell longitudinally is merely that imposed on one-half of it.

9. What is meant by the projected area of a cylinder?

10. Explain why the projected area is taken instead of the circumferential area in determining the force tending to rupture a cylinder longitudinally.

11. State and explain the formula for determining the total internal transverse rupturing force imposed on a cylindrical shell.

12. What is the difference between the above formula and the one used for determining the total stress set up in a longitudinal riveted seam?

13. Are the ends or heads of a boiler considered in determining the resisting strength against transverse pressure? Why?

14. State and explain the formula for determining the safe total resisting strength against transverse pressure.

15. What is meant by a factor of safety?

16. State and explain the meaning of the formula for computing the safe unit transverse pressure which may be imposed on a boiler shell.

17. How is this formula of Question 16 derived?

18. Demonstrate the change that takes place in the formula for determining the safe unit transverse pressure when the radius is used instead of the diameter.

19. If a formula is given by which the *safe* pressure in a shell may be determined, how is the formula altered so that it will give the *bursting* pressure for that shell?

20. State and demonstrate the derivation of the formula for determining the unit transverse bursting pressure of a seamless shell.

21. What is the tendency of the lengthwise component of the internal pressure in a boiler?

22. Explain the formula for determining the total longitudinal rupturing force imposed upon a cylindrical shell.

23. How is the longitudinal resisting strength of a cylindrical boiler determined? State and explain the formula.

24. Write the formula which may be used for computing the safe unit longitudinal pressure which may be imposed on a seamless cylindrical shell. Give its derivation.

25. What is the relation of the unit longitudinal to the unit transverse stress set up in a shell of uniform thickness?

26. How is the unit longitudinal bursting pressure of a seamless shell determined?

27. How may the stresses set up in a spherical shell be determined?

28. What determining factor must be considered in practical design of steam boilers?

29. What is meant by the *efficiency* of a riveted joint?

30. How are the safe working pressures determined for boilers when the efficiency of the riveted joint is known?

31. Give and explain the A.S.M.E. Boiler Code formula for computing the maximum allowable working pressures for power boilers.

PROBLEMS ON DIVISION 9

1. A cylindrical boiler is 20 ft. long and has inside diameter of 4 ft. What is the total transverse force on the boiler when steam pressure is 125 lb. per sq. in.?

2. The total transverse force tending to rupture a boiler is 1,152,000 lb. If the boiler is 24 ft. long and steam pressure is 120 lb. per sq. in. what is the internal diameter?

3. When a boiler drum is 40 ft. long and 42 in. in diameter, what total stress will be imposed on a longitudinal seam when the steam pressure is 90 lb. per sq. in.?

4. If the steel plate in Prob. 1 is $\frac{3}{8}$ in. thick, what will be the safe total transverse force in pounds when a factor of safety of 5 is allowed and steel has a strength of 54,000 lb. per sq. in.?

5. A boiler has a steel sheet $\frac{1}{4}$ in. thick and is 3 ft. across inside. If the steel has a strength of 45,000 lb. per sq. in. and a safety factor of 4.5 is desired, what is the safe transverse pressure when the seam is neglected?

6. Find the total longitudinal force in a boiler shell when the internal diameter is 21 in. and steam pressure is 150 lb. per sq. in.

7. What is the safe longitudinal resisting strength of a tank 21 in. in diameter, $\frac{5}{16}$ in. thick when the ultimate tensile strength of the plate is 50,000 lb. per sq. in. and a factor of safety of 5 is assumed?

8. What is the safe internal pressure with reference to roundabout rupture for the boiler of Prob. 7 when seams are neglected?

9. If the boiler of Prob. 8 is of homogeneous material of a uniform thickness with no seams, what will be the safe pressure with reference to longitudinal rupture?

10. A seamless hollow sphere is made of steel $\frac{1}{8}$ in. thick having a strength of 45,000 lb. per sq. in. If it is 15 in. in diameter and a safety factor of 5 is assumed, what will be the safe internal pressure?

11. What is the safe working steam pressure for a cylindrical boiler having a shell with internal diameter of 36 in., $\frac{3}{8}$ in. thick, and a strength of 50,000 lb per sq. in. if the weakest joint has an efficiency of 80 per cent and a factor of safety of 5 is assumed?

12. What should be the thickness of the boiler shell of Prob. 11 if the pressure is to be 200 lb. per sq. in.?

DIVISION 10

JOINTS

214. Rivets and Fusion Welding Are Used for Fastening Together the Edges of Plates Used for Making Boilers so That the Resulting Pressure Vessels May Be Steamtight.—Until 1931 only riveted joints were permitted but today the trend is more toward the use of fusion welding in which the weld is made with metal in the molten state without the application of mechanical pressure or blows. Welding is used mostly in the construction of drums for water-tube boilers. Manufacturers of fire-tube boilers and smaller water-tube types are using riveted joint principally because they have not found it economical to install the expensive testing equipment required by the Boiler Code when fusion welding is used.

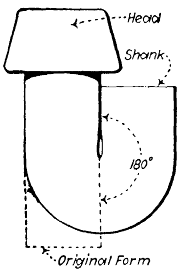


FIG. 156.—Bend-test of a steel or iron rivet.

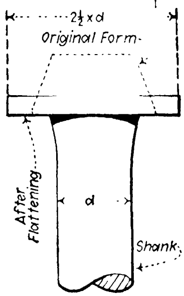


FIG. 157.—Flattening test for steel rivet head.

215. Either Iron or Steel Rivets May Be Used (Table IV).—The tension test (A.S.M.E. Code) for finished steel rivets should be made on a specimen of such length that the elongations may be measured on a gage length of not less than four times the diameter of the rivet. Both iron and steel rivets should endure, without cracking on the outside, being bent upon themselves (Fig. 156) through 180 deg. The head

of a steel rivet should endure without cracking the edges (Fig. 157), being flattened to $2\frac{1}{2}$ times the diameter d of the shank. The heads of iron rivets should withstand being bent back (Fig. 158), thus showing that they are joined firmly to the shank.

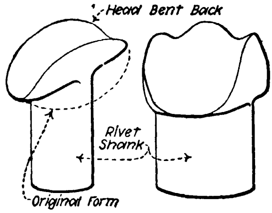


FIG. 158.—Iron rivets with heads bent back without cracking.

216. The Riveted Joint Is, Usually, the Weakest Element of the Pressure Vessel.—Hence it is important that the relative strength or *efficiency* be known. The efficiency of a joint is the ratio, expressed as a percentage, of *the strength of the riveted joint to the strength of the solid boiler plate.* The strength of a riveted joint varies with the form of joint. It usually ranges from about 55 to 95 per cent of that of the plate which is riveted.

Example.—See following Table V for values of the efficiencies of joints of the different types.

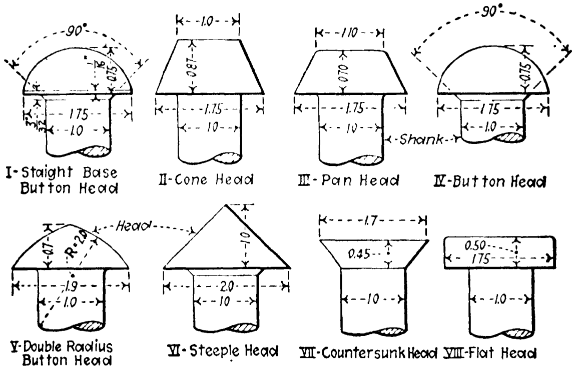


FIG. 159.—Acceptable forms of rivet heads (A.S.M.E. Code, p. 78).

217. Rivet heads of acceptable forms are shown in Fig. 159 which is from the A.S.M.E. Code. The head of a rivet should be so proportioned that the strength against shearing off of the rim is as great as or greater than the tensile strength of the shank.

218. The Holes for Rivets May Be (1) Punched, (2) Drilled, (3) Drilled and Reamed, (4) Punched and Drilled.—If the hole is punched, the material is, especially in thick plate, crushed around the hole and is thereby injured. This injured material must be removed subsequently by reaming or drilling the hole. It is advisable to punch the rivet holes smaller than required and to then, after the plates have been assembled, ream them to size. Thereby it may be insured that the resulting reamed holes in the two or three plates are exactly in line.

NOTE.—The following is from the A.S.M.E. Code, par. 253:

"All holes for rivets or staybolts in plates, butt straps, heads, braces and lugs shall be drilled; or they may be punched at least 1/8 in. less than full diameter for material not over 5/16 in. in thickness and at least 1/4 in. less than full diameter for material over 5/16 in.

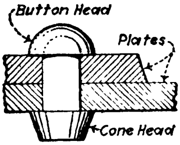


FIG. 160.—Rivet completely fills hole after proper riveting.

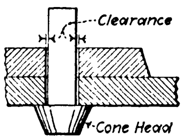


FIG. 161.—Rivet before riveting.

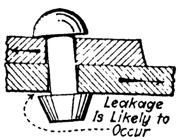


FIG. 162.—Rivet does not fill hole and is not driven tightly.

"Such holes shall not be punched in material more than 5/8 in. in thickness.

"For final drilling or reaming the hole to full diameter, the parts shall be firmly bolted in position by tack bolts.

"The finished holes must be true, clean and concentric."

219. "Rivets shall be of sufficient length to completely fill the rivet holes (Fig. 160) and form heads at least equal in strength to the bodies of the rivets" (A.S.M.E. Code, par. 255). If the rivet is not upset sufficiently to fill the hole a very weak joint will, probably, be the result. Steam and water (Figs. 161 and 162) may leak out around such a rivet.

220. "Rivets shall be machine driven wherever possible, with sufficient pressure to fill the rivet holes and shall be allowed to cool and shrink under pressure" (A.S.M.E. Code, par. 256). There are three methods of driving rivets: (1) by hand, (2) with a pneumatic hammer, (3) by machine. The

latter method is the best because with it the work is uniform and no injury is caused to the surrounding plate. When riveted with a pneumatic hammer, and especially when riveted by hand, the rivet head is often eccentric.

NOTE.—The riveting machine presses the rivet into form and thus holds it for a short time interval, allowing it to cool and contract. The force

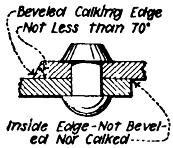


FIG. 163.—The calking edge is beveled.

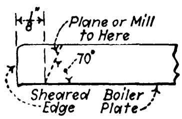


FIG. 164.—Sheared edge of boiler plate should be cut off.

may be from 25 to 150 tons, depending on the size of the rivet which is being driven.

221. "The calking edges of plates, butt straps and heads shall be beveled to an angle not sharper than 70 deg. (Fig. 163) to the plane of the plate, and as near thereto (to 70 deg.) as practicable. Every portion of the sheared surfaces of the

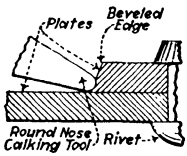


FIG. 165.—No injury to lower plate from round nose calking tool.

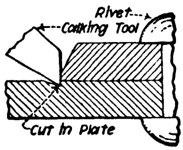


FIG. 166.—Injury from sharp calking tool.

calking edges of plates, butt straps and heads shall be planed, milled or chipped (Fig. 164) to a depth of not less than 1/8 in. When the plate is sheared, the metal along its sheared edge is crushed and distorted. Stresses are set up therein. By planing or otherwise cutting off the sheared edge, the metal thus injured is removed. Calking (Fig. 165) shall be done with a round-nosed tool" (A.S.M.E. Code, par. 257). If a sharp-cornered calking tool (Fig. 166) is used, the lower plate may be scored or cut, thus rendering it weaker. Calking is usually on the outside of the vessel.

welding may also be used to seal the ends of inner butt straps of longitudinal joints.

223. The six possible modes of failure of a riveted joint are: (1) The rivets may shear (Fig. 167). (2) The plate may rupture between the rivets (Fig. 168) due to a tensile stress. (3) The plate or the rivet may crush (Fig. 169). (4) The plate may shear out in front of the rivet (Fig. 170). (5) The plate may tear in front of the rivet (Fig. 171). (6) The failure may be due to a combination of any two or more of the preceding modes. Figures 172 and 173 show how the tensile and com-

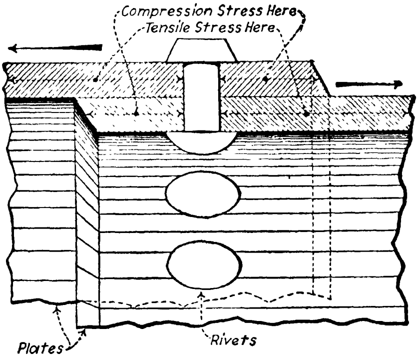


FIG. 173.—Enlarged view showing stresses in seam.

pressive stresses, which tend to cause failure of riveted joints of a boiler, are produced.

224. Riveted joints may be so designed that failure by any one of the six modes above specified cannot occur under normal conditions. For example, the margin may be provided of sufficient width to prevent the tearing (Fig. 171) or the shearing (Fig. 170) of the metal in front of the rivet. In checking a joint for strength, it is unnecessary to consider combinations of failures.

NOTE.—Experience has shown that if a joint is sufficiently strong to withstand failure by Modes (1), (2), and (3), it will also, if designed as directed in Secs. 225 and 226, withstand failure by (4) and (5). Hence in the practical design and checking for strength of riveted joints, only Modes of Failure (1), (2) and (3) need be considered. In following Sec. 242, it is shown hcw the load which will cause failure by Modes (1), (2), or (3) can be determined.

225. Proportions of "back pitch" for riveted joints, which will provide a rational design are specified in the A.S.M.E. Code, par. 182, and are quoted below. These proportions are such that a joint designed in accordance with them will not fail by shearing of the plate (Fig. 170), splitting of the plate in front of the rivet (Fig. 171), or by rupture of the plates between rivets (Fig. 168). Quoting: "The distance between two center lines of any two adjacent rows of rivets, or the

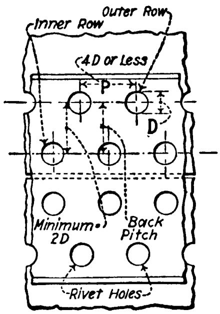


FIG. 174.—Back spacing of rows of rivets when P/D equal 4 or less. (Rivet in the inner row comes midway between two rivets in the outer row.)

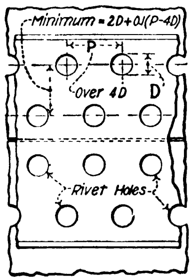


FIG. 175.—Spacing of rivets when P/D is greater than 4. (Rivet in the inner row comes midway between two rivets in the outer row.)

'back pitch' measured at right angles to the direction of the joint shall have the following minimum values:"

(a) If P/D is 4 or less:

$$\text{Minimum back pitch} = 2D \qquad \text{(in.)} \tag{47}$$

(b) if P/D is over 4, then:

$$\text{Minimum back pitch} = 2D + 0.1(P - 4D) \qquad \text{(in.)} \tag{48}$$

where P = pitch of rivets in outer row, *when the rivet in the inner row comes midway between two rivets in the outer row,* in inches. Also, P = pitch of rivets in the outer row less pitch of rivets in the inner row, *when two rivets in the inner row come between two rivets in the outer row,* in inches. (It is here assumed that the joints are of the usual construction where

the rivets are symmetrically spaced.) D = diameter of the rivet holes, in inches.

Example.—See Figs. 174, 175, 176, and 177.

226. "On longitudinal joints, the distance from the centers of rivet holes to the edges of the plates, except holes in the

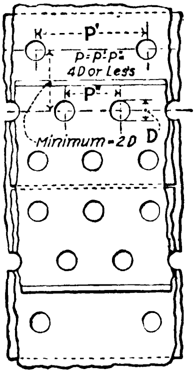


Fig. 176.—Spacing of rows of rivets when P/D equal 4 or less. (Two rivets in the inner row come between two rivets in the outer row.)

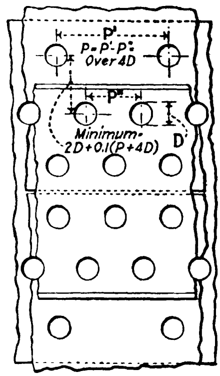


Fig. 177.—Spacing of rivets when P/D is greater than 4. (Two rivets in the inner row come between two rivets in the outer row.)

ends of butt straps, shall be not less than $1\frac{1}{2}$ and not more than $1\frac{3}{4}$ times the diameter of the rivet holes" (A.S.M.E. Code, par. 183). Figure 178 shows the manner of measurement. These proportions insure that the metal in the margin will not, under normal conditions, fail by splitting or shearing out.

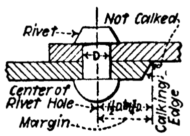


Fig. 178.—Margin of riveted joint.

227. "The strength of circumferential joints of boilers, the heads of which are not stayed by tubes or through braces, shall be sufficient, considering all methods of failure, to resist the total longitudinal force acting on the joint with a factor of safety of five" (A.S.M.E. Code, par. 184*a*). The strength of the longitudinal joint should be determined by the load,

due to the pressure expected in the boiler, which it must withstand. A weaker circumferential than longitudinal joint is permissible owing to the fact that the force tending to rupture the boiler along the circumferential joint is one-half of that tending to rupture it along the longitudinal joint (Sec. 209).

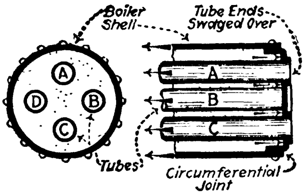


FIG. 179.—Tubes decrease stress in outer shell.

228. The determination of the required strength of the circumferential joint at a head, when tubes acting as stays support it, may be made thus: "When 50 per cent or more of the load which would act on an unstayed solid head of the same diameter of the shell, is relieved by the effect of tubes or through stays, then in consequence of the reduction of the area acted on by the pressure and the holding power of the tubes and stays, the strength of the circumferential joints in the shell shall be at least 70 per cent of that required for the unstayed head joint" (A.S.M.E. Code, par. 184*b*).

Explanation.—In Fig. 179 the area on which steam exerts pressure—the area which is stippled in the illustration—is, obviously less than area on which steam would press if there were no tubes. Not alone do the tubes *A* to *C* decrease the pressure-bearing area, but in addition they assume a portion of the internal longitudinal, thrust against the head, which is due to the contained steam. They assume this thrust because they are headed or riveted over as shown. Thereby the tubes relieve the head of a part of the force which would otherwise be imposed on the outer shell through the circumferential joints. Tubes, when riveted over as shown, brace the heads in about the same manner as do stays.

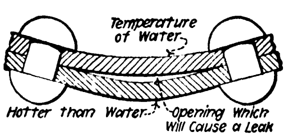


FIG. 180.—Leakage in fire seam when pitch of rivets is too large and allows bulging.

229. When the rivets in a circumferential joint are exposed to the products of combustion, as in a horizontal return-tubular boiler, the shearing strength of the rivets shall not be less than 50 per cent of the full strength of the plate corresponding to its thickness at the joint (A.S.M.E. Code, par. 184*c*). This provision is to insure that each rivet will be safe, even though it becomes very hot owing to poor con-

duction of heat away from its center. To satisfy the above requirement, the rivets must have a comparatively small pitch. Hence the expanding of the outer lap of the joint (Fig. 180) will be minimized. This tends to prevent leaks caused by bulging.

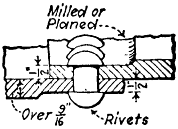


FIG. 181.—Circumferential fire joint for thick plates.

230. When the boiler plate is so thick that it is subject to overheating at a circumferential fire seam, it should be planed or milled down as shown in Fig. 181. This applies to horizontal return-tubular boilers which have plates exceeding $\frac{5}{8}$ in. in thickness. If this cutting away of the plate should cause the joint to be weaker than specified elsewhere, then it should be left thicker at the joint (A.S.M.E. Code, par. 185).

231. Longitudinal Joints May Be of Lap-riveted Construction If the Boiler or Drum Is Not Over 36 In. in Diameter and the Steam Pressure Does Not Exceed 100 Lb. per Sq. In.—If the diameter is greater than 36 in., or the pressure is over 100 lb. per sq. in., the joint should be of butt- and double-strap construction (A.S.M.E. Code, par. 188). The butt joint is the stronger and more dependable, hence this specification.

232. The longitudinal joints of horizontal return-tubular boilers should be located above the fire line of the setting (A.S.M.E. Code, par. 189). When subjected to high temperature and the consequent overheating, the failure of the joint is probable. Hence, for safety, all joints should be so located that they will not be exposed to excessive heat.

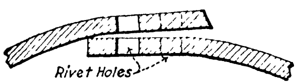


FIG. 182.—Lapped ends of plate as ring comes from rollers.

233. "In Horizontal Return-tubular Boilers with Lap Joints, No Course Shall Be Over 12 Ft. Long.—With butt- and double-strap construction, longitudinal joints of any length may be used, if the steel-plate test specimens are taken lengthwise of the greatest stress, and the other standard tests are satisfied" (A.S.M.E. Code, par. 190).

234. "Butt straps and the ends of shell plates which form the longitudinal joints shall be rolled or formed by pressure,

not by blows, to the proper curvature" (A.S.M.E. Code, par. 191). If the ends of the plate are brought together as shown in Fig. 182 and then hammered into position, the material may be injured.

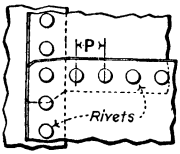


FIG. 183.—Lap joint, single riveted.

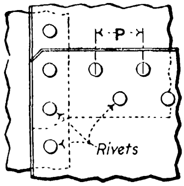


FIG. 184.—Lap joint, double riveted, staggered.

235. A unit strip of a riveted joint is the shortest length, along the joint that divides the rivets symmetrically. It is a strip with width equal to the maximum pitch (Figs. 183, 184, 185, 186, 187, and 188). When the imaginary center lines,

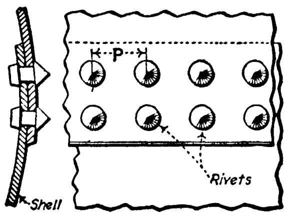


FIG. 185.—Lap joint, double riveted (chain).

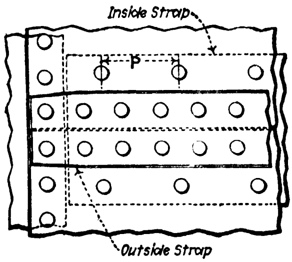


FIG. 186.—Butt joint, double riveted.

which thus divide a joint into unit sections, pass through a rivet, only one-half of the rivet is considered in the calculation.

NOTE.—In computing the strength of a practical boiler joint, the strength for the entire sheet could be estimated. But such procedure would be tedious because of the unnecessarily large number of figures involved in the calculation. The same information concerning the relative strength of the joint may be obtained by considering *any* length of the joint which divides the rivets symmetrically. Hence the "unit strip" is used to conserve time and effort.

236. To compute the tensile strength of a unit strip of solid boiler plate, *multiply together the length of the unit section in inches, the thickness of the plate in inches, and the tensile strength of the material in pounds per square inch.* Or, stating the rule as a formula:

$$\mathbf{S}_T = L_U t S_T \qquad \text{(lb.)} \qquad (49)$$

where $\mathbf{S}_T$ = tensile strength of the unit strip, in pounds,
L_U = width of the assumed unit strip, in inches,
t = thickness of the plate, in inches,
S_T = unit tensile strength of the material, in pounds per square inch.

Example.—See under How to Compute the Efficiency of a Riveted Joint (Sec. 242).

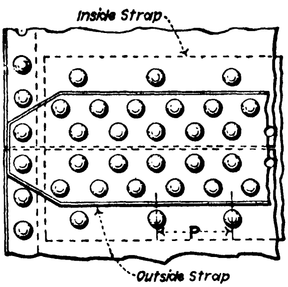


FIG. 187.—Butt joint, triple riveted.

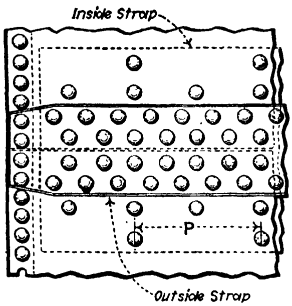


FIG. 188.—Butt joint, quadruple riveted.

237. To compute the tensile strength of the plate between the rivet holes (Fig. 168): *Subtract the diameter of the rivet hole from the width of the unit strip, in inches. Then multiply together the value thus obtained, the thickness of the plate and the tensile strength of the material in pounds per square inch.* Stating these operations as a formula:

$$\mathbf{S'}_T = (L_U - d)tS_T \qquad \text{(lb.)} \qquad (50)$$

where $\mathbf{S'}_T$ = tensile strength of plate between rivet holes, in pounds,
d = diameter of rivet holes, in inches.

The other symbols have the same meanings as specified for the preceding formula.

Example.—See under How to Compute the Efficiency of a Riveted Joint (Sec. 242).

238. To compute the shearing strength of the rivets in a unit strip, *multiply together the area, in square inches, of one rivet to be sheared, the number of rivets, and the shearing strength, in pounds per square inch, of the rivet material.* Expressed as a formula:

$$\mathbf{S}_s = \mathbf{N}AS_s \qquad \text{(lb.)} \qquad (51)$$

where $\mathbf{S}_s$ = shearing strength of the rivets in single shear, in pounds.
$\mathbf{N}$ = number of rivets in unit strip.
A = cross-sectional area, in square inches, of one rivet after driving.
S_s = unit shearing strength of the rivet material, in pounds per square inch.

The above formula will give the *single-shear* (Fig. 167) strength. The *double-shear* (Fig. 189) strength of a rivet may be taken as twice its single-shear strength.

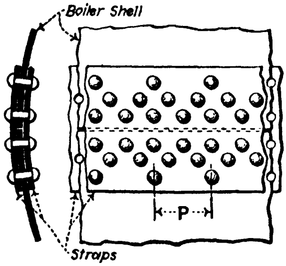


FIG. 189.—Butt joint, triple riveted, straps equal width.

Example.—See under How to Compute the Efficiency of a Riveted Joint (Sec. 242).

239. To compute the crushing strength of the plate in front of the rivets or of the rivets themselves (Fig. 169), *multiply together the number of rivets, the diameter of one rivet in inches, the thickness of the plate in inches, and the crushing strength of the plate material in pounds per square inch.* Expressed as a formula:

$$\mathbf{S}_c = \mathbf{N}dtS_c \qquad \text{(lb.)} \qquad (52)$$

where $\mathbf{S}_c$ = crushing strength of the rivet or plate, in pounds.
S_c = the unit crushing strength of the plate material, in pounds per square inch. The other symbols have the same meanings as specified above.

Example.—See under How to Compute the Efficiency of a Riveted Joint (Sec. 242).

240. The efficiency of a riveted joint is the ratio, expressed as a percentage, of the strength of a unit length of a riveted joint to the strength of the same unit length of the solid plate. It is the percentage strength of the joint as compared with the strength of the plate.

241. The process in determining the efficiency of a riveted joint is to calculate the resisting strength of the joint against failure by: Mode 1, rivets shearing, Mode 2, plate rupturing between rivets, Mode 3, plate or rivet crushing. The weakest mode of failure is taken as the maximum strength of the joint. Then the maximum strength thus determined is divided by the strength of the plate to find the efficiency. The determination of efficiency for a simple joint is illustrated in the following example. The process as applied to joints of the various types is given in detail in the A.S.M.E. Code.

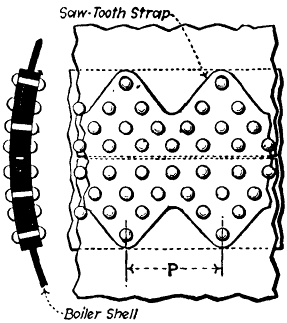


FIG. 190.—Butt joint, quadruple-riveted, saw tooth type.

242. How to compute the efficiency of a riveted joint may be understood from a study of the following example.

Example.—A single-riveted lap joint (Fig. 183) of $\frac{7}{16}$-in. plate has $\frac{7}{8}$-in. rivets with a pitch of 2 in. Tensile strength of plate material is 55,000 lb. Shearing strength of rivet material is 44,000 lb. per sq. in. Crushing strength of plate material is 95,000 lb. per sq. in. What is the efficiency of the joint? *Solution.*—Assume a 2-in. unit strip. The tensile strength of the solid plate, formula (49) $= \mathbf{S}_T = L_U t S_t = 2 \times \frac{7}{16} \times 55{,}000 = 48{,}125$ lb. The tensile strength of the plate between rivet holes, formula (50) $= \mathbf{S}'_T - (L_U - d)tS_t = (2 - \frac{7}{8})\frac{7}{16} \times 55{,}000 = 27{,}070$ lb. The cross-sectional area of a $\frac{7}{8}$-in. rivet is 0.601 in. The shearing strength of one unit in single shear, formula (51) $= \mathbf{S}_s = \mathbf{N}AS_s = I \times 0.601 \times 44{,}000 = 26{,}444$ lb. The crushing strength of the plate in front of one rivet is, formula (52) $= \mathbf{S}_c = \mathbf{N}dtS_c = 1 \times \frac{7}{8} \times \frac{7}{16} \times 95{,}000 = 36{,}367$. Obviously, the joint is weakest in shear. Hence the efficiency of the joint $= \mathbf{E} = \mathbf{S}_s/\mathbf{S}_t = 26{,}444 \div 48{,}125 =$ **0.55 = 55 per cent.**

243. Efficiencies of the various joints vary with the strength of the material that is used in the plates and rivets and with the design of the joint. Conservative values for well-designed joints of good materials are given in the following table.

TABLE V.—EFFICIENCIES OF RIVETED JOINTS

Type of joint	Fig. No.	Efficiency, per cent
Single-riveted lap joint............	183	55
Double-riveted lap joint...........	184, 185	70
Triple-riveted lap joint............	....	75
Single-riveted butt joint...........	....	65
Double-riveted butt joint..........	186	80
Triple-riveted butt joint...........	187	85
Quadruple-riveted.................	188	90
Triple-riveted straps equal width..	189	83
Quadruple-riveted straps saw-tooth.	190	93

NOTE.—The above table lists values for the most commonly used joints. It does not include all joints which may be used. An actual joint may, under test, show efficiencies higher or lower than those quoted, depending upon materials, workmanship, and other variables. Typical joints are shown in Figs. 183 to 190 inclusive. For a complete treatment of the subject of riveted joints, see "Design of Steam Boilers and Pressure Vessels" by Haven and Swett.

244. Disadvantages of Riveted Joints.—Riveted joints for boiler shells and drums have two limitations: one is the fact that the whole shell must be thickened to compensate for their failure to reach 100 per cent efficiency, the other is a tendency toward "caustic embrittlement" at the joints when the boiler water is high in soluble carbonates. Years ago these limitations were behind the demand for legalizing the use of fusion welding in boiler construction. Fear of improper and dangerous welding delayed acceptance by the Boiler Code until 1931 when fusion welding was approved, but surrounded with most rigid safeguards as indicated below.

245. Material used when fusion welding is employed must conform to specifications for steel boiler plate, steel plates of flange, and firebox qualities. The carbon content must not exceed 0.035 per cent.

246. Welded longitudinal and circumferential joints are of the double-welded butt type (Fig. 191) and are reinforced at the center of the weld on each side of the plate by at least $\frac{1}{16}$ in. for plates up to $\frac{5}{8}$ in. thick and by at least $\frac{1}{8}$ in. for heavier plates. This reinforcement may later be removed,

FIG. 191.—Boiler drum showing butt double-welded joint reinforced at center as required by the boiler code.

FIG. 192.—Method of welding head to boiler drum.

but if not removed must be built up uniformly from the surface of the plate to a maximum at the center of the weld. Edges of head and girth joints should be kept separated, at the point of welding, enough to insure thorough penetration of the weld. No unreinforced hole may be located in a welded joint.

247. Stress Relieving.—All fusion-welded drums or shells must be stress relieved. This is done by heating uniformly

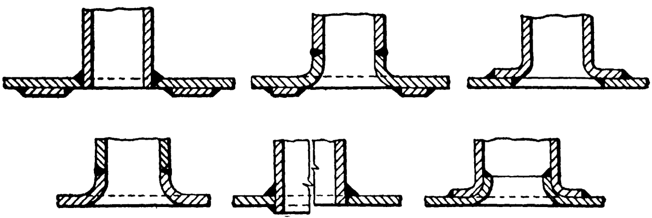


FIG. 193.—These (and many other) methods of welding nozzle connections to boiler drums are acceptable if they meet dimensional and other requirements of the boiler code.

to at least 1100°F., and up to 1200°F. or higher if this can be done without distortion. The drum or shell is held at that temperature for a period proportioned on the basis of 1 hr. per in. of thickness and then allowed to cool slowly in a still atmosphere.

248. Test Specimens.—Two sets of test plates of the same steel as the shell plates may be attached to the shell plate being welded, one set on each end of the joint so that edges to be welded in the test plates are a continuation of correspond-

ing edges of the shell joint. The weld metal is deposited on the test plates continuously with the weld metal deposited in the shell joint. As an alternate method, two sets of detached test plates may be welded. The test plates must not be heated to a temperature higher than that used for stress relieving the drum or shell.

249. Tension Tests.—Two types of tension-test specimens are required, one of the joint and the other of the weld metal. The tension specimen of the joint is taken transverse to the weld and of full thickness of the weld plate after the outer and inner surfaces of the weld have been machined to a plain surface flush with the plate. The tension-test specimen of the weld metal is taken entirely from the deposited weld metal and shall meet the following requirements: Tensile strength shall be at least equal to the minimum tensile strength of the plate which is welded. Minimum elongation shall be 20 per cent in 2 in. For a plate less than $\frac{5}{8}$ in. thick the weld-metal tension test may be omitted.

250. Bend Tests.—Bend-test specimens shall be transverse to the weld. Such a specimen shall be bent cold under free bending conditions until the least elongation measured within or cross the entire weld on the outside fibers of the bend-test specimen is 30 per cent When a crack is observed on a convex surface of the specimen on the edges the specimen is considered to have failed. Cracks at the corners of the specimen are not to be considered a failure.

251. Specific Gravity of Weld Metal.—Specific-gravity specimens taken from the weld metal shall when possible be 2 in. long and $\frac{5}{8}$ in. diameter, the minimum specific gravity shall be 7.80.

252. Nondestructive Tests.—For wall thickness of $4\frac{1}{4}$ in. and less every portion of all longitudinal- and circumferential-welded joints must be examined by X ray or gamma ray. Acceptability of welds examined by X ray are judged by comparing the X-ray film with a standard set which may be obtained from the boiler code committee.

253. Joint Efficiency.—When calculating the maximum allowable working pressure the efficiency of welded joints may be taken as 90 per cent.

QUESTIONS ON DIVISION 10

1. For what is a riveted joint used?

2. Of what material are rivets made?

3. What are the tests for rivets?

4. What part of a pressure vessel is weakest?

5. What is meant by the efficiency of a riveted joint?

6. How should the holes be made and be prepared for rivets? Why?

7. Why should a rivet fill the hole after being driven?

8. What are the three methods of driving rivets? Which method is the best?

9. Describe the calking edge of a plate.

10. Discuss the six modes of failure of a riveted joint.

11. Is it possible to eliminate the probability of failure by any of these modes of failure? How?

12. What are the formulas for determining the minimum back pitch?

13. Why is the distance from the edge of the plate, in longitudinal joints, to the center of the rivet specified? What is the specification?

14. What should be the strength of the circumferential joint as compared to the longitudinal joint of a boiler?

15. How is the calculation for the circumferential joint at the head of a drum calculated when tubes act as stays?

16. When a boiler plate is over $^9/_{16}$ in. thick, or so thick that it is subject to overheating, at the circumferential joint what should be done?

17. When the boiler drum is not over 36 in. in diameter and the pressure is not to be over 100 lb., what kind of a joint may be used for the longitudinal seam?

18. Why should a seam in a boiler drum not be exposed to the high temperature of the furnace?

19. What is the limit of length of joint allowed by the A.S.M.E. Code in a horizontal return-tubular boiler when the joint is of the butt and double-strap construction?

20. How shall the plates that lap to make the longitudinal joint be formed to curvature? Why should they not be hammered to shape?

21. State and explain the formula for computing the strength of a unit strip of solid boiler plate.

22. State and explain the formula used for computing the tensile strength of a riveted unit strip of boiler plate.

23. How is the shearing strength of the rivets in a unit strip determined?

24. State and explain the formula for computing the crushing strength of the plate in a unit strip.

25. How is the maximum strength of a unit strip determined?

26. How is the efficiency of a joint determined?

27. What is the range of the efficiencies of the joints of the various types?

28. How are welded joints reinforced?

29. Describe stress relieving.

30. To what tests must welds be subjected?

31. What joint efficiency is used when welding is employed?

32. Is it permissible to weld boiler drums without submitting the weld to X-ray examination?

PROBLEMS ON DIVISION 10

1. A double-riveted, lap joint (Fig. 184) is riveted with steel rivets $\frac{3}{4}$ in. in diameter. The pitch of the rivets is $2\frac{1}{2}$ in. and the thickness of the plate is $\frac{5}{16}$ in. What is the efficiency of the joint?

DIVISION 11

BRACES AND STAYS

254. The tendency of a pressure within an approximately cylindrical vessel is to force the shell into a truly cylindrical shape and to force the heads into truly hemispherical shape. Hence any pressure-sustaining boiler surface, which is neither hemispherical nor cylindrical in shape, has forces imposed on it which tends to cause the surface to assume one of those shapes.

255. When a vessel has a flat pressure-confining surface of relatively great area, it is necessary to stay that surface to prevent distortion and the possible rupture which might occur owing to the tendencies described in the paragraph just preceding. Since cylindrical, spherical, and hemispherical surfaces are "self-staying" (because of the reasons outlined in Div. 9), they require no bracing.

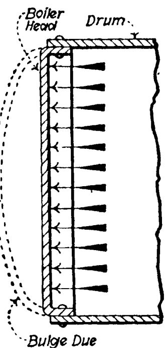


FIG. 194.—Internal pressure tends to bulge head.

Example.—If a flat boiler head (Fig. 194) is subjected to a high internal pressure, its tendency is to assume the bulged shape indicated by the dashed outline. To prevent the plate from assuming the bulged shape, and possibly to prevent it from rupturing, stays or braces must be used to hold it.

256. Boiler Stays Should Satisfy Three Requirements.—(1) The stays should be of sufficient number and strength that they will—assuming that the plate itself has no strength nor stiffness—wholly support the plate. (2) The stays should be so disposed that they will offer minimum interference to inspection of the boiler. (3) The stays should be so arranged they will offer minimum obstruction to the circulation of water in the boiler.

257. Boiler stays may be divided into three general classes: (1) through stays, (2) diagonal stays, (3) gusset stays. Each is treated in following sections. Through stays are not desirable for spans exceeding 20 ft. in length. If used for longer spans, they will sag in the middle and not assume the total load imposed on the end plates.

NOTE.—In general, through stays are preferable, structurally, to diagonal stays. Sometimes it is necessary to apply either through or diagonal stays to provide a construction which will not interfere with the inspection of the interior of the boiler. Gusset stays usually interfere materially with inspection and impede water circulation. Hence they are now seldom used in America.

258. The procedure in designing the staying for a flat surface is: (1) Compute the force of the pressure against the surface which is to be stayed; (2) space and design the stays so that they will safely support this force. When tubes pass through a plate they will support part of the surface.

259. How to design the staying for the end plate of a horizontal return-tubular boiler (Fig. 195), in accordance with A.S.M.E. Code requirements, will now be explained: It is assumed that the tubes support the lower part, $ACNM$, of the sheet and that the shell supports a strip ABC of the width L_w which is determined thus:

$$L_w = \frac{5t}{\sqrt{P}} \qquad \text{(in.)} \tag{53}$$

where L_w = width of the strip which may be assumed to be supported by the boiler shell, in inches.

t = thickness of the plate in the head, in *sixteenths* of an inch.

P = maximum allowable pressure in pounds per square inch.

It is assumed the 2-in. wide area. ADC, above a line just outside of the tubes, is supported by the tubes. Now, the area of the segment $ABCD$ which remains to be stayed, should be determined. This area may be ascertained directly from tables, which will be found in various engineer's handbooks. Or it may be computed by applying formula (54) given in the following section. The area of the segment having

been determined, and the maximum allowable steam pressure being known, the total force which will be exerted against the segment is found by multiplying *the steam pressure per square inch* by *the number of square inches in the segment.* The

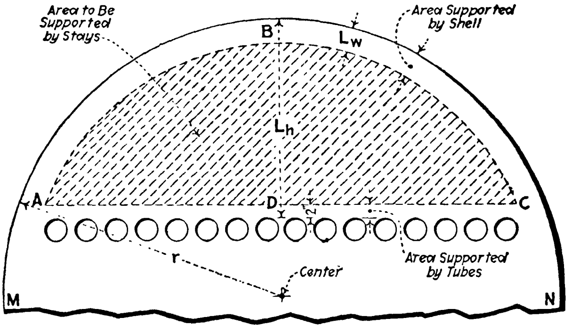


FIG. 195.—Determining net area of segment of boiler head for staying (A.S.M.E. Code p. 55).

spacing of the stays is then ascertained by applying formula (56) given in a following section. It is now only necessary to compute a value for the diameter of the stays. This should be such that when the stays jointly assume the total

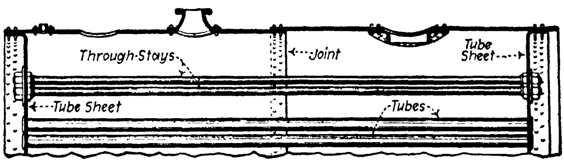


FIG. 196.—Through stays in a Scotch boiler.

force imposed on the segment, the allowable stress in them will not be exceeded.

260. To compute the area of a segment of a circle the following formula (see Fig. 195) may be used:

$$A = \frac{4(L_h - L_w - 2)^2}{3}\sqrt{\frac{2(r - L_w)}{(L_h - L_w - 2)} - 0.608} \qquad \text{(area, sq. in.)} \quad (54)$$

where A = the area of the segment to be stayed, in square inches.

r = the radius, (Fig. 195) of the shell, in inches.
L_h and L_w = the distances, in inches, as shown in Fig. 195.

261. Diagonal Stays Must Be Designed to Assume Greater Tension Than Corresponding Stays Perpendicular to the Stayed Surface.—If the stays are through stays perpendicular to the surface to be supported (Fig. 196), the total resistance offered by the stays should be equal to the total pressure imposed on the segment to be supported. But if they are

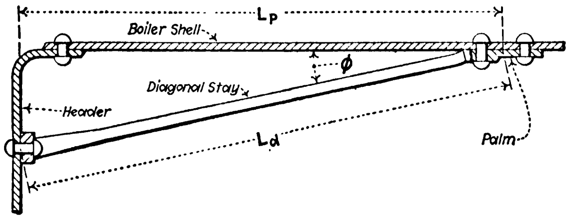


FIG. 197.—Stress in diagonal stay (A.S.M.E. Code, p. 59).

diagonal stays (Fig. 197) the cross-sectional area of each may be calculated by applying this formula:

$$A_d = \frac{A_t L_d}{L_P} \qquad \text{(sq. in.)} \qquad (55)$$

where A_d = the cross-sectional area of the diagonal stay, in square inches.
A_t = the cross-sectional area of a through stay of the same metal, which would safely support the load, in square inches.
L_d = length of the diagonal stay, in inches.
L_P = distance from the surface supported to the center of the palm of the diagonal stay, in inches.

Explanation.—Formula (55) follows from the law of the resolution of forces. If line AO (Fig. 198) be drawn proportional in length to the stress in the through stay, then line BO will be proportional in length to the stress in the corresponding diagonal stay. Hence $A_d{:}L_d{::}A_t{:}L_P$, therefore, $A_d = A_tL_d/L_P$.

262. The formula for computing the proper spacing of stays in flat plates is given in the A.S.M.E. Code, par. 199:

$$L_p = t\sqrt{C \div P} \qquad \text{(in.)} \qquad (56)$$

where L_p = distance (Fig. 199) between stay centers, or the pitch, in inches.

t = thickness of the stayed plate, in *sixteenths* of an inch.

C = a constant, values for which are given in the following table.

P = maximum allowable working pressure, in pounds per square inch.

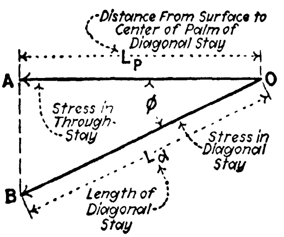


FIG. 198.—Illustrating the derivation of the diagonal-stay formula.

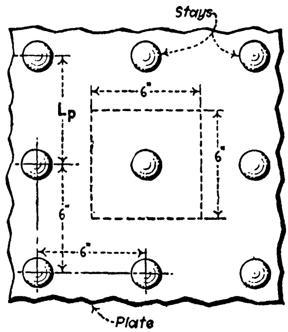


FIG. 199.—Stay supports area 6 × 6 in.

For usual conditions the pitch of the stays, as determined by the formula, will be from 5 to 7 in.

TABLE VI.—VALUES OF C FOR THE STAY-SPACING FORMULA

Condition	Value of C
Stays screwed through plates not over 7/16 in. thick and riveted	112
Stays screwed through plates over 7/16 in. thick and riveted..	120
Stays screwed through plates and fitted with nuts outside or with inside and outside nuts omitting washers...........	135
Stays fitted with heads not less than 1.3 times the diameter of the stays, screwed through the plates or made a taper fit and having the heads formed on the stays before installing them and not riveted over, said heads being made to have a true bearing on the plate............................	150
Stays fitted with inside and outside nuts and outside washers where the diameter of the washers is not less than 0.4 L_p and the thickness not less than t......................	175

Example.—Assuming that, for given conditions, a 6-in. pitch is found to provide the proper spacing. Then each stay (Fig. 199) supports

6 $\times$ 6 = 36 sq. in. of surface. If the pressure is to be 150 lb. per sq. in., then a through stay should be of such diameter that it will safely sustain a load of 36 $\times$ 150 = 5,400 lb. Strictly, the cross-sectional area of the stay should be deducted from the unit area to be supported before figuring the load to be sustained.

263. Maximum Stresses Allowable in Stays and Stay Bolts. Because of the excessive stresses and corrosion to which stay bolts may be subjected, the allowable stresses which are specified by the A.S.M.E. Boiler Code (par. 220*c*) and which are here tabulated are very low. Stay bolts may be of either iron or steel.

TABLE VII.—MAXIMUM ALLOWABLE STRESS IN STAYS

Class	Description of stays	Allowable stresses, lb. per sq. in.	
		For lengths between supports not exceeding 120 diameters	For lengths between supports exceeding 120 diameters
a	Unwelded or flexible stays less than 20 diameters long, screwed through plates, with ends riveted over..................	7,500	
b	Hollow steel stays less than 20 diameters long, screwed through plates, with ends riveted over...	8,000	
c	Unwelded stays and unwelded portions of welded stays, except as specified in class *a* and class *b*....	9,500	8,500
d	Steel through stays exceeding 1½-in. diameters	10,400	9,000
e	Welded portions of stays..	6,000	6,000

264. Through stays are stay rods which connect one end of the boiler with the other (Fig. 196), thus supporting both

end plates so that they will not bulge. The ends of the stays may be fastened to the plates as shown in Fig. 200.

265. Diagonal stays are made in many different forms. Figure 201 illustrates typical designs. An installation is shown in Fig. 38. In Fig. 202 is illustrated the *crowfoot diagonal stay* Another form of fastener for the end of a crowfoot stay is delineated in Fig. 203.

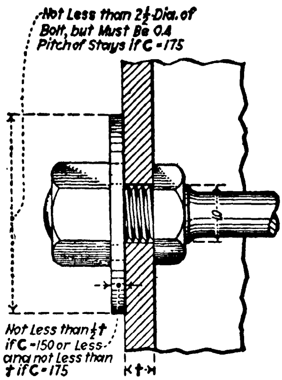


FIG. 200.—Through-stay end.

266. Girder stays (Fig. 204) are used for supporting crown plates or other flat plates which may bound a firebox. That of Fig. 204 is in one piece. Other types are illustrated in Figs. 33 and 34.

267. Sling stays (Fig. 205) are also used to support crown sheets. A sling stay may extend from the boiler shell to a girder stay as indicated in Fig. 205.

268. Gusset stays (Fig. 206) perform the same function as do diagonal stays. They are made of flat plate and are much

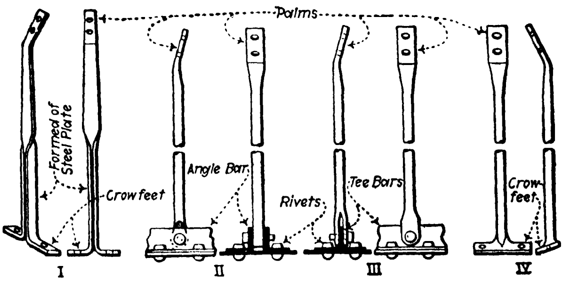


FIG. 201.—Diagonal stays.

stiffer than the stays of the other types. Hence great care must be exercised in their installation. If improperly set, excessive expansion and contraction stress may result.

269. Stay Bolts Are Short "Through Stays."—They are used in bracing water-leg plates and other flat surfaces which are relatively close together. They are applied most frequently

in locomotive and marine-type boilers as shown in Figs. 33

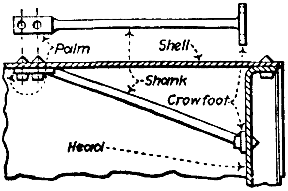


FIG. 202.—Crowfoot stay (riveted feet).

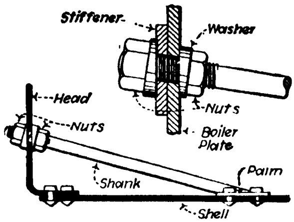


FIG. 203.—Form of fastening for diagonal stays in Scotch boilers.

and 34. Ordinarily, the bolt (Figs. 207 and 208) is threaded, screwed in place, and riveted over. In Fig. 208 is shown a bolt which has the threads turned off to minimize corrosion. Corrosion, so experience shows, is exceedingly pronounced at the root of the thread. The end of a stay bolt (Fig. 209) may be secured by nuts.

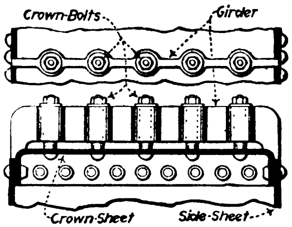


FIG. 204.—One piece girder stay.

270. Stay Bolts Are Subjected to Bending as Well as to Tensile Stresses.—The bending stresses are due to the unequal expansion of the plate near the fire and that heated only by the water. This bending stress increases

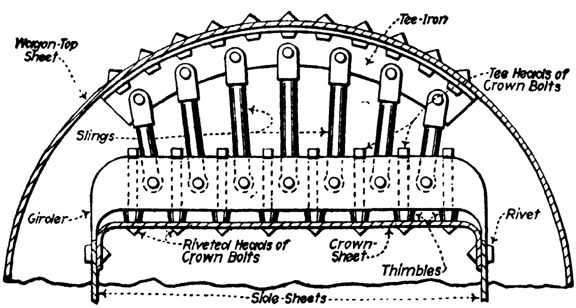


FIG. 205.—Sling stays supporting girder stay.

as temperature differences occur. Finally, the bolt may break near the plate without the rupture being apparent from the out-

side. A hole drilled in the end of the bolt (Fig. 208) permits steam or liquid to escape as soon as the rupture penetrates

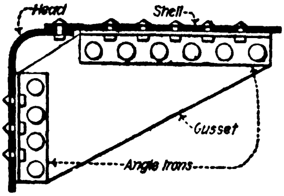


FIG. 206.—Gusset stay.

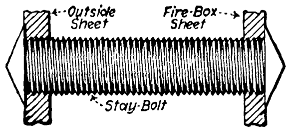


FIG. 207.—Stay bolt.

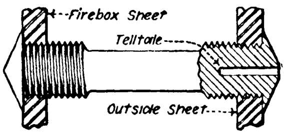


FIG. 208.—Stay bolt.

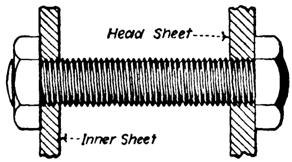


FIG. 209.—Stay bolts with nut.

to the center of the bolt. Thus an automatic "tell-tale" is provided. To allow for lateral movement and thus minimize bending stresses, flexible stay bolts (Fig. 210) have been designed. They have not been and probably will not be employed generally because of their high cost, their complication, and of the relatively large space which they occupy.

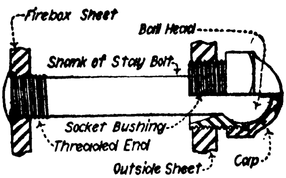


FIG. 210.—Flexible stay bolt.

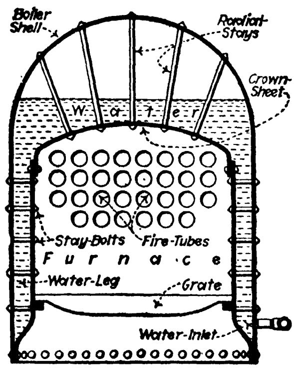


FIG. 211.—Radial stays in a locomotive type boiler.

271. Radial stays (Fig. 211) are employed to stay two plates which have different radii of curvature. They are used principally in locomotive boilers.

272. Steel Angles May Be Used for Staying Tube Sheets (Fig. 212).—The A.S.M.E. Code permits such staying, where

the boiler does not exceed 36 in. in diameter and the pressure is not greater than 100 lb. per sq. in. It also specifies the sizes of the angles which may be so used.

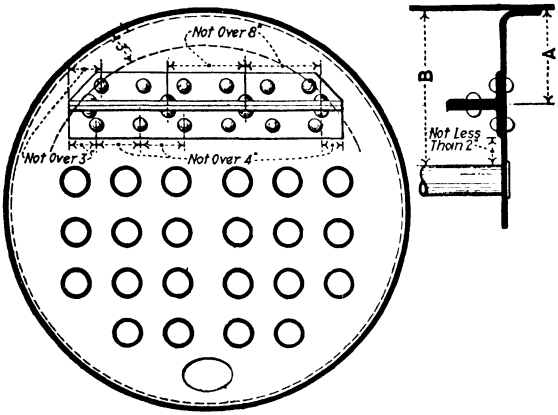


FIG. 212.—Staying of head with steel angles (A.S.M.E. Code, page 62).

QUESTIONS ON DIVISION 11

1. What is the effect of applying pressure inside of a vessel when it is deformed from a truly cylindrical form?
2. Why is it necessary to stay a flat surface?
3. What three requirements should a boiler stay satisfy?
4. What are the three general classes of stays?
5. What is the procedure for designing stays?
6. How is the width of the strip adjacent to the shell in a horizontal return-tubular boiler, which is assumed to be supported by the shell, determined?
7. State and explain the use of the formula for computing the area of a segment.
8. How is the area of a diagonal stay determined?
9. Give the formula for finding the spacing of stays in flat plates.
10. Why are the allowed stresses in stays much lower than the ordinary safe load for the material?
11. What are through stays? Diagonal stays? Girder stays? Sling stays? Gusset stays?
12. What is a stay bolt? Where are they used?
13. What is the manner of preparing a stay bolt so that it may be known when it cracks due to repeated bending?
14. When are radial stays used?
15. When are steel angles permitted for use for staying a flat plate?

DIVISION 12

FIRE TUBES AND WATER TUBES

273. The purpose of fire tubes and water tubes in boiler construction is to increase the evaporative efficiencies of the boilers above those possible with a plain-cylinder boiler. Introduction of fire tubes and water tubes accomplishes this by increasing the ratio of heating area to volume of the water space.

274. Fire tubes afford internal passages, within a boiler shell, through which the gas of combustion flows from the combustion chamber (Figs. 27 and 28) or firebox (Figs. 36 and 42) to the smokebox. They are surrounded by water.

275. Water tubes divide the water space of a boiler into a number of sections. The space within each tube constitutes a section. Water circulates through the tubes while the hot gas surrounds them externally.

276. The most important stress in a water tube is the transverse stress which tends to rupture the tube longitudinally. This stress is exactly like that which is exerted by the steam pressure directly against the boiler shell (Div. 9).

277. The stresses in a fire tube or flue are produced as follows: (1) by the pressure of the steam, which is transmitted through the mass of water to the surface of the tube or flue. This produces a transverse crushing stress. (2) By the pressure of the steam (Fig. 213) against the head sheets or tube sheets into which the ends of the tube or flue are secured. This produces a longitudinal tensile stress.

NOTE.—The tensile stresses in fire tubes and flues are, ordinarily, very inconsiderable by contrast with the crushing stresses. Fire tubes and flues perform a subordinate function in staying the sheets (Fig. 213) to which their ends are secured. Failure on account of a pressure acting parallel to the length of a fire tube occurs, therefore, by separation of the tube from the sheet rather than by rupture of the tube itself.

278. Failure of a fire tube or flue under a transverse crushing stress invariably occurs by collapse (Fig. 214) of

the wall of the tube or flue. Theoretically, a fire tube or flue should not fail by application of a uniform pressure against the entire outside area of its wall, until the crushing strength of the tube material is exceeded. This condition would be realized if the tube were ideally cylindrical in contour. But the ideally perfect cylinder is unattainable in practice. The steam pressure tends constantly to augment any slight irregularity that exists in the tube contour. As the distortion develops, resistance to its further increase progressively diminishes. Consequently, the tube or flue will fail by

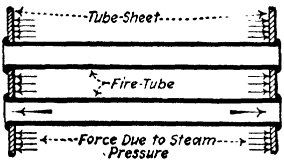


Fig. 213.—Longitudinal stress in fire tube.

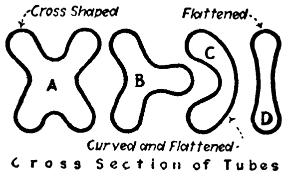


Fig. 214.—Collapse of tubes due to external pressure.

collapse long before the material becomes stressed to the limit of its compressive strength.

279. The diameters and thicknesses of water tubes for various pressures are given in Table VIII. The greater the diameter and the greater the steam pressure, the greater must be the tube thickness.

280. The A.S.M.E. Boiler-code minimum thickness of tubes used in fire-tube boilers measured by Birmingham wire gage, for maximum allowable working pressures exceeding 175 lb. per sq. in., is as in Table IX.

281. Tube holes in drums or headers may be drilled full size or they may be punched ½ in. smaller in diameter and then drilled, reamed, or finished full size with a rotary cutter. The internal diameter of finished holes shall not exceed the nominal diameter of the tube by more than the following amounts:

	Inches
Fire end of fire-tube boiler tubes	1/32
Opposite end of fire-tube boiler tubes	1/16
Water tubes	1/32

TABLE VIII.—A.S.M.E. BOILER CODE MAXIMUM ALLOWABLE WORKING PRESSURE FOR SEAMLESS AND LAP-WELDED STEEL TUBES FOR WATER-TUBE BOILERS FOR DIFFERENT DIAMETERS AND GAGES OF TUBES

Outside diameter of tube, in in. D	Thickness of tube in Birmingham wire gage and in inches												
	17 $t = 0.058$	16 $t = 0.065$	15 $t = 0.072$	14 $t = 0.083$	13 $t = 0.095$	12 $t = 0.109$	11 $t = 0.120$	10 $t = 0.134$	9 $t = 0.148$	8 $t = 0.165$	7 $t = 01.80$	6 $t = 0.203$	5 $t = 0.220$
½	434	686	938	1,334									
¾	206	374	542	806	1,094								
1	...	218	344	542	758	1,010							
1⅛	...	166	278	454	646	870	1,046						
1¼	...	124	225	383	557	758	916	1,118					
1½	...	...	146	278	422	590	722	390	1,058				
1¾	...	...	...	203	326	470	583	727	871	1,046			
2	...	...	...	146	254	380	479	605	731	884	1,019		
2¼	...	...	...	.....	198	310	398	510	622	758	878	1,062	
2½	...	...	...	....	153	254	333	434	535	657	765	931	1,053
2¾	...	...	...	....	117	208	280	372	464	575	673	824	935
3	..	...	...	.....	.....	170	236	320	404	506	596	734	836
3¼	...	...	..	....	.....	.....	199	276	354	448	531	658	752
3½	...	...	...	.....	.....	...	167	238	310	398	475	594	681
3¾	...	...	...	....	....	....	139	206	273	355	427	537	619
4	...	...	...	.....	....	....	.....	178	240	317	385	488	565
4½	...	...	...	.....	.....	....	.....	.....	186	254	314	406	474
5	...	...	...	...	.....	....	...	..	142	204	258	340	402

$P = \left(\frac{t - 0.039}{D}\right)18000 - 250$ where P = maximum allowable working pressure, in lb. per sq. in. t = thickness of tube wall, in in. D = outside diameter of tube, in in.

NOTE.—Maximum allowable working pressures for superheater tubes shall be the same as for boiler tubes.

The A.S.M.E. Boiler Code, par. 250, requires that a fire-tube boiler shall have both ends of the tubes substantially rolled and beaded or the tubes may be rolled, beaded, and

TABLE IX.—GAGE THICKNESS OF WALLS OF STEEL OR WROUGHT-IRON TUBES OR FLUES FOR FIRE-TUBE BOILERS, B.W.G.

Outside diameter, in.	Maximum allowable working pressure, lb. per sq. in.				
	175	200	225	250	275
1	13	12	12	12	12
1½	13	12	12	12	12
1¾	13	12	12	12	12
2	13	12	12	12	12
2¼	13	12	12	12	11
2½	12	11	11	11	10
3	12	11	11	10	10
3¼	11	10	10	9	9
3½	11	10	10	9	9
4	10	9	9	8	8
4½	10	9	8	8	7
5	9	8	7	7	6
5½	9	8	7	6	6
6	8	7	6	5	5

NOTE.—The rule followed in the above table is that, for pressures above 175 lb. per sq. in., the maximum allowable working pressure is increased for each additional gage in thickness by 200 lb. per sq. in. divided by the diameter of the tube in inches. This rule may be followed for determining values intermediate of those given in the table and for values beyond the limits of the table. For pressures below 175 lb. per sq. in., the gage thickness shall be the same as for 175 lb. per sq. in.

welded around the edge of the head. Projecting tube ends are prohibited for the reason that the heat of combustion will burn off such projections. Paragraph 251, requires that the ends of all tubes, suspension tubes, and nipples in water-tube boilers and superheaters shall be flared not less than 0.125 in. over the diameter of the tube hole. Or, otherwise, the tube may be flared not less than 0.125 in., and then rolled and beaded or flared rolled and welded.

282. Defective fire or water tubes may be removed from a boiler by one of the following operations: (1) By cutting a

lengthwise slot in each end of the defective tube with a cape chisel (Fig. 215) and then battering in (Fig. 216) the metal adjacent to the edges of the slot. The tube can then be drawn out through the tube hole in the sheet which is located most conveniently for the manipulation. This method is applicable for water tubes, or where the end of a fire tube has not been upset with a beading tool. (2) By first cutting off, with a flat chisel, the beaded end of the tube flush with the tube sheet, and then slotting and battering in the end

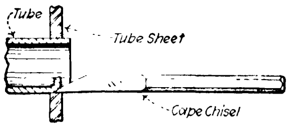


FIG. 215.—Cutting tube with cape-chisel.

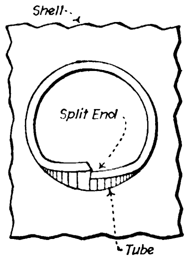


FIG. 216.—Tube bent in preparation for removal.

as explained above. (3) By cutting off the end of the tube just inside the sheet. This requires the use of a special cutting tool.

283. The procedure of replacing a fire or water tube in a boiler is as follows:

(1) The tube holes should be examined to determine their condition. If found nicked or otherwise damaged, they should be turned up with a reamer. If the edges of the tube holes

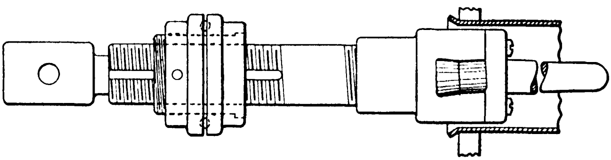


FIG. 217.—Roller-tube expander to roll and flare in one operation for use in water-leg boilers. (*A. L. Henderson's Sons.*)

are sharp, they should be blunted with a half-round file. The new tube should now be placed in position, with its ends projecting through the sheets from 0.25 in. to 0.5 in. (A.S.M.E. Boiler Code, par. 252).

(2) If a water tube is being replaced the ends should preferably be rolled and flared with an expander similar to

that of Fig. 217 or 218. Some times the tube hole will be found somewhat larger in diameter than the tube. In such cases it is necessary to line up the hole. The liners are

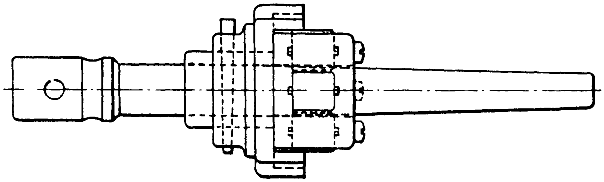


FIG. 218.—Roller expander for use where expander may pass hand holes. (*Richard Dudgeon Inc.*)

preferably made of strips of charcoal wrought iron. But in the absence of wrought iron, soft sheel steet may be used.

(3) If a fire tube is being replaced, the ends may be made secure with an expander (Fig. 218). With this tool only the portion of the tube within the hole is rolled out. If damage to a tube hole has necessitated its enlargement

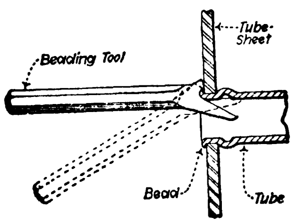


FIG. 219.—Beading tool.

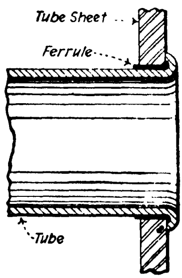


FIG. 220.—Tube with ferrule.

by reaming, the end of the tube may be correspondingly enlarged by brazing on a *copper ferrule* (Fig. 220). The wall of the tube is expanded to a steamtight fit in the hole. Coincidentally, the projection of the tube outside the tube sheet is flared or belled out. The ends are now beaded over (Fig. 219). The dotted lines (Fig. 219) show the position of the beading tool when the work of turning the bead is begun. As the work progresses, the beading tool is gradually brought to the position indicated by the full lines.

QUESTIONS ON DIVISION 12

1. What is the chief advantage of the tubular form of construction in steam boilers?

2. State the distinction between fire tubes and water tubes.

3. What is the principal stress in water tubes?

4. What stresses must fire tubes withstand?

5. Why do failures of fire tubes occur by collapse of the wall of the tube rather than by simultaneous crushing of the material at all points?

6. How are defective tubes removed from a boiler? How replaced?

7. How should the ends of a water tube be secured?

8. What is the purpose of the copper ferrules on the ends of a fire tube?

9. What are the requirements of the A.S.M.E. Boiler Code with respect to the ends of fire tubes and water tubes?

DIVISION 13

MANHOLES AND HANDHOLES

284. Manholes and handholes are necessary openings in steam boilers. They are essential in the building, repairing, and inspection of such pressure vessels.

285. Manholes afford access of a man to the boiler interior. In manufacture, it is necessary in certain riveting operations that a helper be inside of the boiler. When inspecting, the boiler inspector must examine the interior of the shell. This, where the space in the boiler is sufficiently large, he can do more effectively by entering the shell than by merely looking at it through a small hole. Also, it is necessary for a man to enter the shells of the larger boilers for the removal of scale formations. The A.S.M.E. Code specifies certain provisions for manholes, which are abstracted in succeeding sections.

286. Handholes are provided in small boilers and in sectional-leader straight-tube boilers. These handholes are sufficiently large to accommodate a tube expander. They are used in repairing and in inspection, when it is necessary to see inside, and for removing and inserting tubes and other parts.

287. The proper number of and locations for manholes, handholes, and washout holes in a steam boiler are specified in pars. 264 to 266 of the A.S.M.E. Code. These "access" holes should always be so located that they will permit of proper inspection, cleaning, and repairing and yet not affect materially the strength of the boiler structure. The holes should be in positions where highly heated gas will not contact with them.

288. The size of a manhole should be such as will permit an average-sized man to pass easily through it. The size and shape of a manhole may, particularly in the smaller boilers, be of various designs. Generally, however, elliptical openings are provided. Openings of other forms may be necessary owing to a certain arrangement of tubes or other parts.

NOTE.—The A.S.M.E. code, par. 258, provides that the minimum allowable dimensions for an elliptical manhole are, in the clear, an 11-in. minor diameter and a 15-in. major diameter, or it may be 10 by 16 in. When the manhole is circular it should be not less than 15 in. in diameter.

289. Either Wrought Steel, Wrought Iron, or Cast Steel Should Be Used in Manhole Construction.—These materials

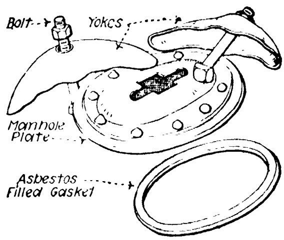


FIG. 221.—Details of forged-steel manhole cover.

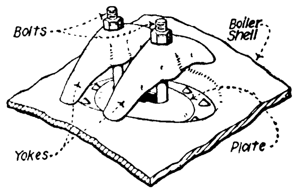


FIG. 222.—Manhole plate in position.

are much stronger and more reliable than the cast iron which formerly was widely employed. Covers or "plates" (Fig. 221) are now of wrought or cast steel. Steel yokes and bolts (Fig. 222) hold the cover in place in the manhole. Gaskets (Fig. 221), which are discussed further in a succeeding section, are always necessary under the covers. The yokes—either one or two may be used—are placed parallel to the minor axis.

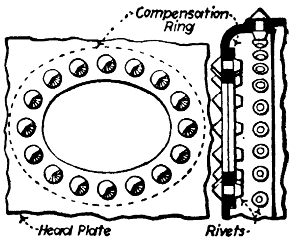


FIG. 223.—Flat strengthening ring.

290. Every manhole opening must be reinforced to compensate for the metal which has been cut away in making the hole. If not thus reinforced, the boiler shell will not have sufficient strength in the vicinity of the hole. A reinforcing method which was formerly used for both head (Fig. 223) and shell manholes involved the riveting on of plain *compensation rings*. These were usually attached on the outside of the shell, but were often placed inside.

NOTE.—Reinforcing or compensation rings were formerly of cast iron. But this metal, because of its inherent weakness and brittleness, was relatively ineffective. Its use for such service is now prohibited by boiler codes.

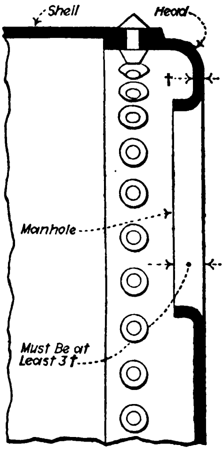


FIG. 224.—Manhole with flanged edges.

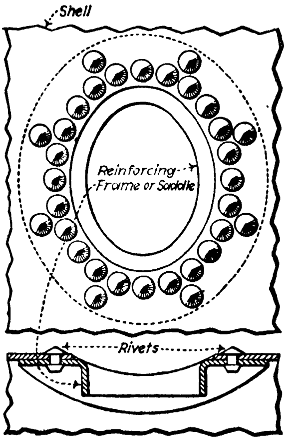


FIG. 225.—Reinforcing frame or saddle.

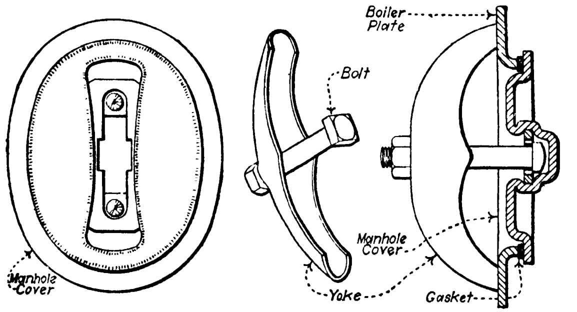


FIG. 226.—Pressed-steel manhole cover and assembly to boiler plate.

291. Modern practice in reinforcing holes in heads is to flange over (Fig. 224) the manhole edges. Since the head plate is stiffened by the flange, the making of a manhole in this manner does not materially weaken the head.

292. In reinforcing manhole openings in drums and shells (Fig. 225), flanged frames or saddles are employed. These

are riveted to the shell plates. When the shell diameter exceeds 48 in., they are double riveted. Such saddles are stamped from hot flat plate. Reinforcing rings may also be fusion welded.

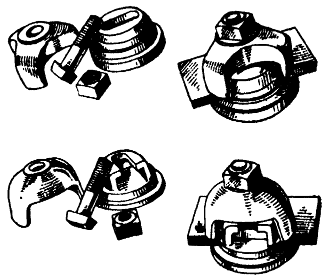


FIG. 227.—Round handhole plate and parts.

293. Gaskets must be used around manholes to prevent leakage. The gasket seat must be so designed as to prevent the gasket from being blown out by the steam pressure. Gaskets should be of a material which will withstand both heat and dampness. Rubber or asbestos compositions are used. Corrugated copper rings are also employed. Gaskets may not be more than $\frac{1}{4}$ in. thick. See Fig. 226 for details of a

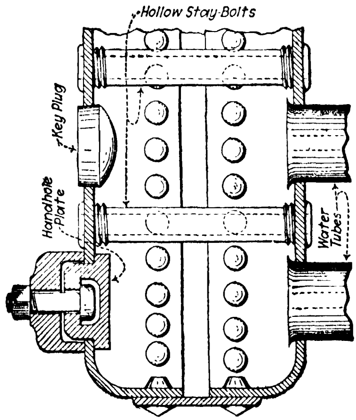


FIG. 228.—Section of lower part of water-leg, showing ordinary and also key handhole plugs in place.

manhole and cover assembly. Width of bearing surface for a manhole gasket should not be less than $\frac{11}{16}$ in.

294. The construction of handholes and handhole plates is similar to that for manholes. Reinforcing rings are not used. The usual form for handholes is elliptical. The average

dimensions are about 3 in. by $4\frac{1}{2}$ in. When located in water legs and headers round and odd-shaped handholes are often used.

295. A round handhole is often used in water legs when the tubes are perpendicular to the leg, as shown in Fig. 227. In Fig. 228 is shown an assembly of a round handhole plate and parts in a water leg.

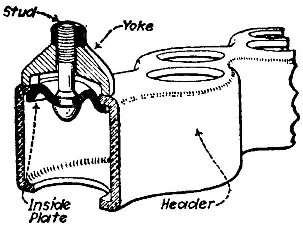


FIG. 229.—Fittings for elliptical hand-hole in wrought steel header.

296. Plates of Elliptical or Elongated Handholes May Be Removed through Their Own Openings.—In Fig. 229 is illustrated a handhole cover which is used in headers where the tube intersects the header at an oblique angle. This plate may be removed from within the opening which it covers. In boilers where the shape of handholes and their covers is such that the cover can not be removed through the hole, then the covers must be removed through a larger handhole at some other location in the boiler or water leg.

QUESTIONS ON DIVISION 13

1. For what are handholes and manholes used?

2. In general, where should manholes and handholes be located?

3. What is the minimum size of a manhole?

4. Of what material should manhole parts be made? Why not cast iron?

5. Why and how much should a manhole be reinforced?

6. What is the method of strengthening the opening in a boiler-drum head?

7. Why are gaskets necessary? Of what may they be made?

8. What is the minimum width of the bearing surface for a gasket?

9. In general, what may be said concerning the construction of handhole openings?

10. How are handhole plates or covers attached?

11. How are handhole covers removed from water legs when they are attached on the inside?

DIVISION 14

BOILER ACCESSORIES

297. Boiler accessories may be defined as appliances, fittings or mountings which are either intimately connected with the boiler structure or with the work of boiler operation and maintenance. They are indispensable to safety, economy, and convenience.

NOTE.—This division deals with the most intimately connected accessories and appliances. See author's "Power-plant Auxiliaries and Accessories" for other power-plant equipment.

298. A safety valve is a device for relieving the pressure in a boiler, or other closed vessel, by allowing the enclosed fluid to escape when the pressure becomes greater than desired. Figure 230 shows a simple, but impracticable, arrangement which would allow the steam to escape if the pressure in the boiler became great enough to push the plate from its seat.

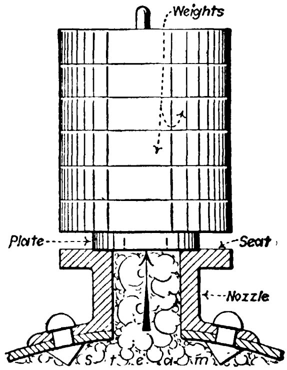


FIG. 230.—Elementary form of safety valve with flat-seated valve.

NOTE.—The safety valve is an insurance against explosion from overpressure.

299. Spring-loaded pop safety valves with seat and bearing surface of the disk inclined at an angle between 45 and 90 deg. to the valve stem are permitted by the A.S.M.E. Boiler Code. Dead-weight safety valves and weighted lever valves are no longer permitted.

Explanation.—A spring-loaded valve with an ordinary flat or bevel-seated disk would be practically useless as a safety valve. When the

steam pressure increased sufficiently to lift the disk against the tension of the spring, a slight opening would occur and a small quantity of steam would escape. The disk would continue to rise in response to a constant increase of pressure. But the closing force of the spring would be constantly augmenting. Consequently, the valve might not give sufficient opening for escape of the steam as fast as the steam would be generated. The pressure might then increase to 20 to 40 lb. greater than the initial

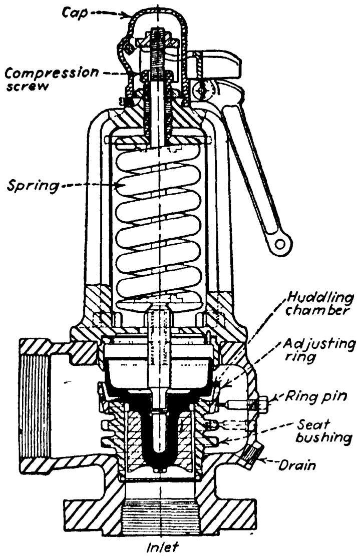


FIG. 231.—Pop safety valve with huddling chamber. (*Consolidated Ashcroft Hancock.*)

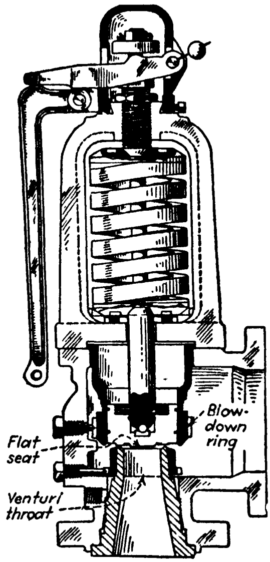


FIG. 232.—Nozzle safety valve for saturated steam. (*Crosby Steam Gage and Valve Company.*)

opening pressure. Therefore, safety-valve seats must be specially designed, as will be shown.

300. The design of a spring-loaded safety valve should be such that the disk will lift instantly to a maximum height when the steam has attained the pressure for which the valve is set. This is accomplished (Fig. 231) by increasing the area against which the steam acts, and hence the opening force, simultaneously with the first slight opening of the valve. The surfaces which enclose the path of the escaping steam may also be so arranged as to utilize the reactionary force of the steam for lifting the valve disk as in Fig. 232.

Explanation.—When the steam pressure against the exposed surface of the valve disk (Fig. 231) is sufficient to cause the disk to rise from its seat a small quantity of steam escapes through the very small opening between the disk and seat. This steam impinges against the top surface of the huddling chamber and is deflected downward. It strikes the bottom surface of the huddling chamber. Thence it passes out through the restricted opening between the lip of the disk and the adjusting ring. This restriction causes a pressure to develop in the huddling chamber This is in addition to the force exerted by the fast-moving steam particles which are escaping through the opening between the disk and seat. Thus the upward pressure against the disk is greatly augmented. The valve spring is compressed accordingly. A sudden wide opening of the valve results. In Figs. 232 and 233 there is no huddling chamber. The valve is initially lifted by the action just described. As the valve lift increases the blowdown ring turns the steam flow downward, which produces a reaction force on the disk causing it to lift to its maximum.

301. The loss of pressure between the popping pressure of a spring-loaded safety valve and the closing pressure is called the *blowback* or *blowdown.* The blowdown allowed is not less than 2 per cent or more than 4 per cent of the pop pressure. Blowdown is adjusted by raising or lowering the adjusting or blowdown ring. In Figs. 231 and 232 the ring pin is removed and a screw driver inserted in the hole engaging the notches in the adjusting ring. In Fig. 232 pushing notches to the right raises the ring and shortens the blowdown. Pushing notches to the left lowers the ring and increases the blowdown. If the adjusting ring is raised too high in attempting to shorten blowdown the opening left may be very low and closing will be indistinct and dragged out. The adjusting ring must be lowered until clean pops and blowdown are obtained. Never move the adjusting ring more than ten notches without retesting.

302. The valve seat and valve disk should be of material which will not corrode and so impair the working of the valve. If these parts were of iron they might become rusted together so that the valve would not open as it should. These parts should be made of monel metal, bronze, nickel, or stainless steel.

303. For computing the discharge capacity of a safety valve see the method used in the Appendix of the A.S.M.E. Boiler Code.

304. Comparison of lifts, discharge areas and relieving capacities of seven different makes of pop safety valves is given in Table X. Diameter of all valves, 4 in.; popping pressure for all, 200 lb. per sq. in. (Revised from Mark's "Handbook.")

TABLE X.—COMPARISON OF SAFETY-VALVE CAPACITIES

Lifts		Effective area of discharge with opening lift, sq. in.	Per cent of largest opening	Relieving capacities	
Opening, in.	Closing, in.			Lb. of steam per hr. $W = 110LDP$.	Hp. on basis of 30 lb. evaporating per hr. = 1 hp.
0.064	0.024	0.568	46 6	6,050	200
0.031*	0.017	0.390	31.4	5,820	195
0.056	0.032	0.496	40.8	5,300	177
0.094	0.039	0.834	68.5	8,910	298
0.094	0.055	0.834	68.5	8,910	298
0.082	0.054	0.727	59.7	7,750	260
0.137	0.088	1.220	100.0	13,000	434

* Flat seat, constant used in capacity formula 155; all other valves with 45-deg. seats.

305. Chattering of Pop Safety Valves Is Generally Due to Incorrect Mounting.—If the safety valve is installed on the end of a vertical pipe of considerable length, or on an elbow or tee attached to a horizontal pipe trouble is sure to result. When the valve opens the frictional resistance of the connections cause a pressure drop. If this pressure drop equals or is greater than the blowdown, the valve immediately closes. This produces a destructive hammering of the valve on its seat. Improper setting of the blowdown ring or use of a valve designed for different pressure will also cause chattering.

306. Safety valves are required on superheaters to protect the superheater tubes from overheating in case steam flow from the boiler is stopped by sudden closing of an engine stop valve. Figure 233 shows a superheater safety valve for high temperature service. The closing spring is further removed from the valve so it will not be effected by high temperature. It is important in purchasing such valves to

specify the steam temperature. Superheater valves should be set to blow before valves on the boiler drum to insure flow of steam through the superheater.

307. When More Than One Safety Valve Is Required for a Boiler, the Valves May Be Mounted on a Single Base.—Figure 235 illustrates such an arrangement for two valves on a Y-base. Area of the connection to the boiler must at least equal the sum of the inlet areas of each valve.

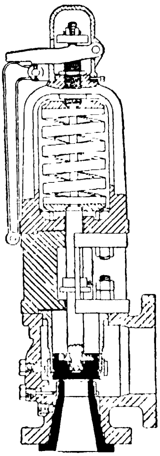


FIG. 233.—Nozzle type super heater safety valve. (*Crosby Steam Gage and Valve Company.*)

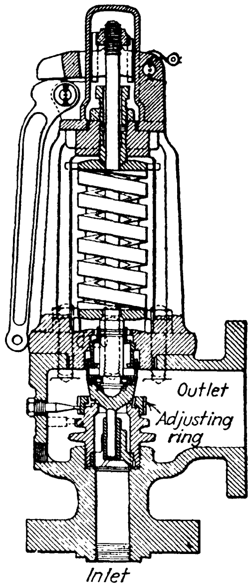


FIG. 234.—Economizer relief valve. (*Consolidated Ashcraft Hancock.*)

308. The proper installation of a pop safety valve requires that it be attached to a nozzle, and as close to the boiler as possible. If piping is used between the boiler and the valve, it should be of a larger size than the nominal diameter of the valve. Care should be taken that no chips, scale, red lead or other substances are left in the inlet of the valve, or in the boiler connections to it.

NOTE.—The first time pressure is raised in a boiler on which new pop valves have been installed, the valve should be opened by pulling the lever when the pressure is within about 5 or 10 lb. of the set pressure stamped on the valve. The valve should be held open about one minute,

or long enough to make sure that all foreign matter has been blown out of the valve and connections. If piping is installed in the outlet of the valve, it should under no circumstances be reduced in size. If more than one fitting is used in the line the entire installation beyond the first fitting should be increased in size. Such piping should be well supported. Improper support of the outlet pipe may result in leakage of the valve on account of vibration. A pop valve should not be installed in a horizontal position. Discharge piping must be adequately drained.

309. The methods of testing installed safety valves are specified and described in A.S.M.E. Code, pars. 275 and 391.

310. An accumulation test for checking the relieving capacity of the safety-valve equipment of a boiler is made by

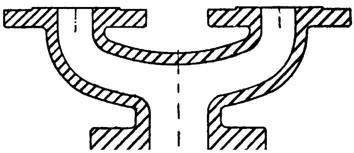


FIG. 235.—Y nozzle for two safety valves.

forcing the fire under the boiler to the limit of furnace capacity while all outlets, except the safety valve or valves, are closed against escape of steam from the boiler. In this test the safety-valve equipment must show a relieving capacity (A.S.M.E. Code) sufficient to prevent the steam pressure from rising more than 6 per cent above maximum allowable working pressure or more than 6 per cent above the highest pressure at which any value is set.

NOTE.—An accumulation test should not be made if the rate of firing cannot conveniently be increased very gradually, or if the conditions are such that the intensity of the fire cannot be kept under complete control. In a test of this kind provision should be made for liberating the steam through an auxiliary outlet that can be used in case the safety valve fails to work properly. For a boiler under which coal is burned with natural draft, the auxiliary outlet should have an area of not less than one square inch for each two square feet of grate surface.

311. In Operating a Boiler, Raising the Steam Pressure to the Blowing-off Point, Except as a Test Precaution, Should Be Avoided.—Considerations of economy dictate this, inasmuch as the steam discharged through a safety valve represents

a measurable loss of heat energy and money. The following example is illustrative.

Example.—A 3-in. 45-deg. beveled pop safety valve blows during an average period of 4 min. in each hour of a 24-hr. run. The average lift of the valve is 0.09 in. The blowback is from 100 to 97 lb. per sq. in. gage. What is the resulting expense if the coal costs $4.00 per ton, and 8 lb. of water are evaporated for each pound of coal burned? *Solution.*—Formula from A.S.M.E. Code, par. A-11: $W = 110PDL$, where, W = weight of steam discharged per hr., in lb., P = average absolute pressure of steam, in lb. per sq. in., D = inside diameter of valve seat, in in., L = vertical lift of valve disk, in in. The average absolute pressure = $(100 + 97) \div 2 + 14.7 = 113.2$ lb. per sq. in. Substituting in formula: $W = 110 \times P \times D \times L = 110 \times 113.2 \times 3 \times 0.09 = 3362$ lb. of steam per hr. Minutes of total discharge per day = $4 \times 24 = 96$ min. or 1.6 hr. Steam discharged in 24-hr. run = $3362 \times 1.6 = 5379$ lb. Coal required to evaporate = $5379 \div 8 = 672$ lb. Cost = $672 \div 2000 \times 4.00$ = $1.34 per day; or in a month of 30 days: 30×1.34 = $40.20.

312. Safety-valve requirements and specifications are divided by the A.S.M.E. Code into three divisions as determined by the character of the installation: (1) power boilers, new installations; (2) power boilers, existing installations; (3) heating boilers, new installations.

313. Some of the important safety-valve specifications for new installations are given in brief below. These are from the A.S.M.E. Code, par. P-269.

1. When the relieving capacity must be over 2,000 lb. per hr. use two safety valves.
2. The relieving capacity must be great enough that the pressure in the boiler will never rise over 6 per cent above the maximum allowed boiler pressure, or 6 per cent above the highest pressure setting of any valve.
3. One or more valves should be set at or below the maximum allowable working pressure. Remaining valves may be set within a range of 3 per cent above allowable working pressure.
4. Weighted-lever safety valves shall *not* be used.
5. Relieving capacities are to be based on: (1) 6 lb. of steam per hr. per sq. ft. of boiler heating surface for water-tube boilers. (2) 5 lb. for all other boilers with maximum allowable working pressures over 100 lb. per sq. in. (3) 3 lb. for 100 lb. per sq. in. or less. Superheating surface is not included.
6. No other valve of any description shall be used with a safety valve. A safety valve should be located as close as possible to the boiler.

7. When a muffler is used, it shall not cause any back pressure or interference with discharge.

8. A lifting device is required for raising the seat $\frac{1}{16}$ in. when no pressure is on the boiler.

9. Safety-valve springs shall not be used for pressures over 10 per cent above or below that for which they are designed.

10. Every superheater shall have one or more safety valves near the outlet.

11. Valves discharging superheated steam shall have steel bodies, valve seats of nickel composition or equivalent, and the spring outside

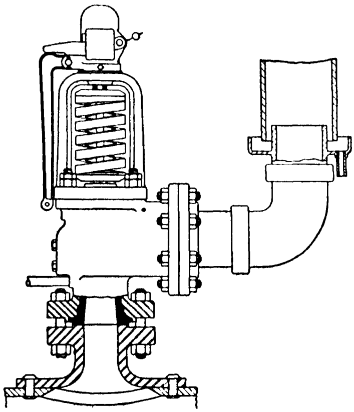


FIG. 236.—Recommended safety-valve installation. Note slip joint in discharge piping to relieve valve of pipe strain.

the valve. They should be set to blow before valves on the boiler drum to insure flow of steam in superheater.

314. Safety valves for steam heating boilers shall be pop safety valves of the spring-loaded type which cannot be set for a pressure greater than 15 lb. per sq. in. No valve should be less than 1 in. or greater than $4\frac{1}{2}$ in. standard pipe size. The sizes are determined from the grate area as indicated in a table in the Code. Other requirements are similar to those above specified in connection with safety-valve installations for power boilers.

315. Feed-water Inlets Are Necessary for All Steam Boilers. Since the water in the boiler is converted to steam and passes out as such, feed water must be supplied for replenishment. The feed water may be forced into a boiler by a pump, by

an injector, or other device. Feed piping must carry the water from the feeding apparatus to the boiler. Inlets, valves, and other required appurtenances must be so arranged as to provide maximum effectiveness.

316. The Location of the Feed-water Inlet of a Boiler Is Important.—The following are determining factors: (1) the tendency of the comparatively cool feed water to set up stresses due to unequal expansion in the boiler plate and tubes;

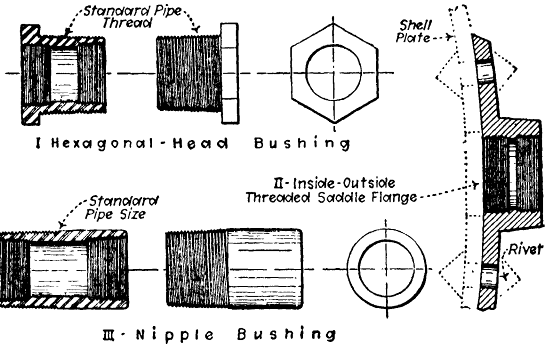


FIG. 237.—Boiler bushings and flange (A.S.M.E. Code).

(2) liability of the boiler plate to deteriorate over an area near the discharge orifice.

317. The feed pipe of a horizontal return-tubular boiler for power purposes should (A.S.M.E. Code) enter the front head immediately above the tubes and should extend back about $\frac{2}{5}$ of the length of the boiler. The water should be discharged above the central rows of tubes. By flowing through a length of pipe inside the boiler, the feed water becomes heated to a comparatively high temperature. The piping should be securely fastened above the tubes. The attachment of the feed pipe to the head of the boiler should be made with a bushing (Fig. 237) or flange.

NOTE.—In boilers of all types which employ an internal extension of the feed pipe, a boiler bushing (Fig. 237), or its equivalent, must be used for attaching the feed pipe to the head or shell. (A.S.M.E. Code, par. 315.)

318. Boiler Feed Water Should Not Be Discharged Near a Riveted Joint or Furnace Sheet (A.S.M.E. Code).—Feed

water is always cooler than boiler water. The comparatively cool feed water would chill and contract the plate or seam upon which it might be permitted to discharge. The local contraction thus produced would cause severe straining of the metal. A permanent weakening tendency would also develop in the material of the plate or seam. Baffles or other means must be provided to prevent this.

319. Feed water may be discharged in the form of a spray by allowing it to spill over the edge of a pan (Figs. 238 and 239). The *spray pan* may be a long trough with saw-toothed edges.

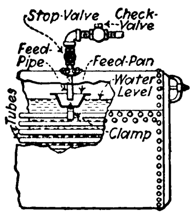


FIG. 238.—Top feed and spray pan in horizontal return-tubular boiler.

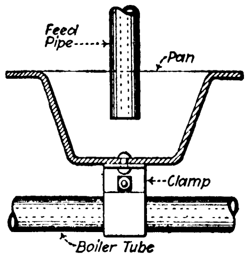


FIG. 239.—Detail of spray pan.

This method is objectionable on account of a tendency of the sprayed feed water to entrain with the outgoing steam current. Such entrainment may be obviated by locating the spraying device as far as possible from the steam connection.

320. Feed Water Should Be Discharged in That Region of the Water Space Where Ebullition Is Least Violent.—Precipitation of the impurities in the mud drum or adjacent to the blowoff orifice is thus facilitated.

321. A Feed Pipe Shall Be Provided with a Check Valve Near the Boiler and a Valve or Cock between the Check Valve and the Boiler.—When two or more boilers (A and *B* Fig. 240) are fed from a common source, there shall also be a globe valve *G* on the branch to each boiler, between the check valve *C* and the source of supply *S*. Whenever globe valves are used on feed piping, the inlet shall be under the disk of the valve. (A.S.M.E. Code, par. P-317.)

322. A Check Valve Is Designed to Permit Flow in One Direction Only (Fig. 241).—These valves restrain automatically the back flow of water, from the boiler through the feed pipe, when the pump or injector is stopped. Valves of certain

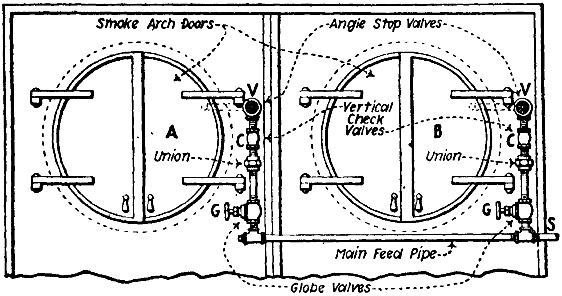


FIG. 240.—Feed pipe connections to two horizontal return tubular boilers.

types may be installed either horizontally or vertically in the line, while others may be used only in a horizontal line.

323. The stop valve or cock, between the check valve and the boiler (V, Fig. 240) is used to isolate the check valve from the boiler for inspection or repair. The stop valves G between the check valve and the source of supply is for regulating the supply or cutting it off altogether.

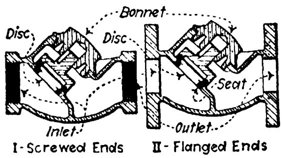


FIG. 241.—Check valves.

324. The size of pipe required for boiler-feed lines is determined in accordance with the velocity of flow through the pipe. A pipe size is selected that will limit velocity to between 480 and 600 ft. per min.

325. Two Stop Valves Should Be Placed in the Steam Connection Leading from Each Boiler in a Battery, to the Steam Main.—This is required by the A.S.M.E. Code when two or more boilers are connected to a steam main. One of these valves should preferably be an automatic stop valve.

326. Automatic non-return stop valves in the main steam connections of boilers are necessary (Figs. 242, 243, and 244) for the following principal reasons: (1) They provide a means for automatically cutting a boiler from the main

line in the event of tube failure, blowing out of a manhole gasket, rupture of a blowoff connection, or similar accident. (2) They render certain that no steam from the main line will be admitted to a cold boiler when there are workmen inside of it. (3) They provide a means whereby a boiler may be

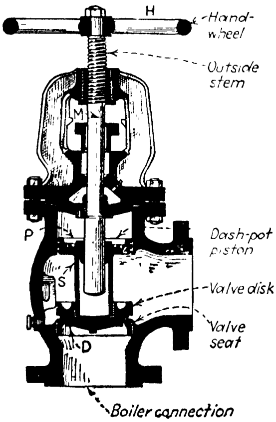


FIG. 242.—Edward single-acting automatic stop valve for 400-lb. pressure.

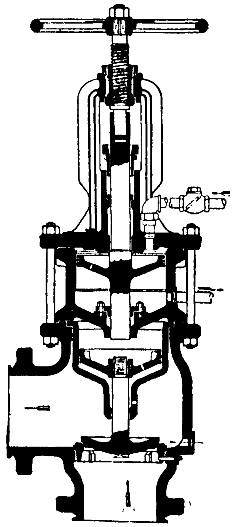


FIG. 243.—Double-acting automatic boiler stop valve. (*Schutte and Koerting Company.*)

automatically cut into service when its pressure is equal to the line pressure.

327. Automatic stop valves for use in the main steam connection of boilers may be divided into two classes: (1) single-acting non-return stop valves which close automatically when steam begins to flow from the main header into the boiler; (2) double-acting stop valves which close automatically either when an abnormal flow of steam from a boiler begins, or when steam begins to flow from the main header into the boiler.

328. The function of a single-acting automatic stop valve (Fig. 242) is to stop a flow of steam from the main header

into the boiler in case any part of the boiler is ruptured. Also, it automatically cuts the boiler in and out of service. When the pressure in the boiler, becomes as great as that in the main header, the valve opens automatically.

Explanation.—The main working part of the valve (Fig. 242) is a disk *D* which is seated as in an ordinary stop valve. The disk is connected by a stem *S* which has a dashpot piston *P* on its upper end. The dashpot is designed to cushion the opening and closing of the valve. If the disk were not thus restrained, it would chatter and hammer itself to destruction. When the steam pressure under the disk becomes a little greater than that above it, the valve opens. It has a tendency to remain open as long as steam flows upward. When the pressure below the valve diminishes and the steam begins to flow downward, the valve closes immediately. The valve may be closed definitely by screwing down the stem *M* with the handwheel *H*. Thus it is evident that an automatic non-return stop valve is essentially a dash-pot-cushioned check valve provided with a screw for keeping it closed when required. It performs a function in the branch steam line somewhat similar to that of the check valve in the water-feed line.

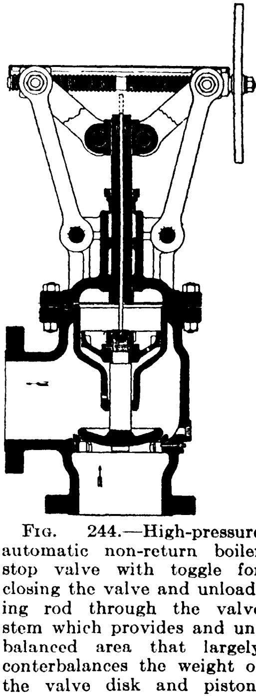


Fig. 244.—High-pressure automatic non-return boiler stop valve with toggle for closing the valve and unloading rod through the valve stem which provides and unbalanced area that largely conterbalances the weight of the valve disk and piston. (*Schutte and Koerting Company.*)

329. The function of a double-acting automatic stop valve (Fig. 243) is threefold: (1) It isolates the boiler from the main header, by closing against the header pressure in the event of rupture of any part of the boiler. (2) It isolates the boiler from the main header, by closing against the boiler pressure, in the event of rupture of the main header or of apparatus which is supplied with steam from the main header. (3) It automatically cuts the boiler in and out of service under normal operating conditions.

330. Automatic Stop Valves May Increase the Economy of a Plant.—It is uneconomical to run underloaded boilers.

They should be cut in and out of service in conformity with the load requirements. With ordinary stop valves alone on the connections, this duty is neglected in most cases on account of the effort and inconvenience involved in opening and closing the valves. But with automatic valves in working order, the task is reduced to a simple matter of banking the fires under the unnecessary boilers when the load drops off and leveling them again when the load increases. The automatic valves will respond to the changed conditions. Without such valves deadened, boilers may be permitted to float along on a line and so become a source of loss, due to condensing of the steam which passes in from the main header.

331. An Ordinary Stop Valve Should Supplement the Automatic Stop Valve on the Main Steam Connection of a Boiler.—It should be inserted in the line between the main header and the automatic valve. The automatic valve should be as close to the boiler as possible. The most important reason for putting in the supplementary valve—which should be a gate valve—is to provide a means for shutting off the line pressure from the automatic valve when the boiler is laid up for cleaning or repairs. At this time the automatic valve should be inspected so that no conjecture may exist as to its working condition. It should be thoroughly cleaned and all rubbing surfaces lubricated sparingly with graphite. All bearing surfaces should be free from abrasions or scratches which might prevent free action.

NOTE.—Steam- and drain-pipe connections for the bodies of automatic non-return stop valves are described in a following section.

332. A steam gage is an instrument for indicating the steam pressure in a boiler or other containing vessel. Ordinarily, a steam gage registers the pressure of the steam, above atmospheric pressure, in pounds per square inch.

333. Modern Steam Gages Are Generally of the Bourdon, or Spring-tube Type.—These gages operate through the tendency of a curved tube (Figs. 245 and 246), which is of oval cross section, to straighten out when pressure is applied internally. The spring tubes are made of brass or composition metal.

NOTE.—For a discussion of the principles of steam-gage operation see the Author's "Practical Heat."

334. Every Boiler Should Have a Steam Gage.—The gage may be connected to the steam space, the water-column,

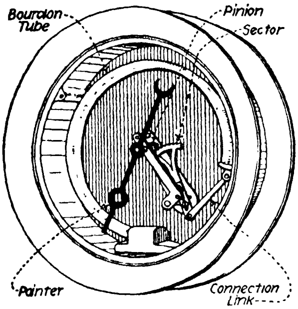


FIG. 245.—Interior view of steam gage with single Bourdon tube. (*Ashton Valve Co.*)

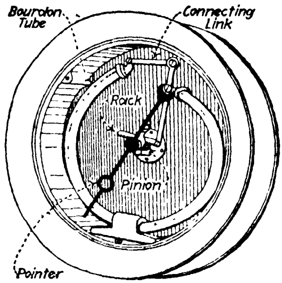


FIG. 246.—Interior of steam gage with double Bourdon tube. (*Ashton Valve Co.*)

or to the steam connection to the water column. Each gage must be connected to a siphon (Figs. 247, 249, and 250) or equivalent device with sufficient water capacity to prevent the steam from entering the gage tube. This is to obviate the deteriorating and disturbing effect of the high temperature of the steam on the material and mechanism of the gage.

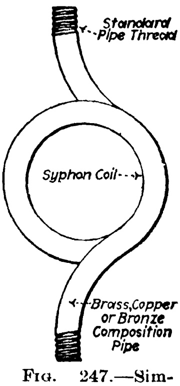


FIG. 247.—Simple form of siphon for steam gage connection.

335. It Should Not Be Possible to Shut

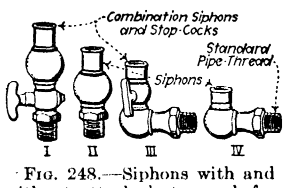


FIG. 248.—Siphons with and without attached stop-cock for vertical and horizontal connection to steam gage.

Off the Gage from the Boiler, except by a Cock Placed Near the Gage.—Exception to this rule may be made when there

is a long connection between the gage and the boiler. In such a case a valve or cock may be placed near the boiler, but it must be of a type that can be locked or sealed open. The handle of the cock (Figs. 248 and 250) should be parallel to the pipe in which it is located when the cock is open. Connections to gages should be of brass, copper, or bronze composition (A.S.M.E. Code, Par. 296).

336. The Dial of a Steam Gage Should Be Graduated to Not Less than 1½ Times the Maximum Allowable Steam

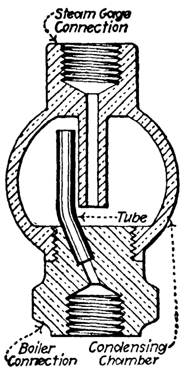


FIG. 249.—Sectional elevation of ball siphon and steam-gage connection.

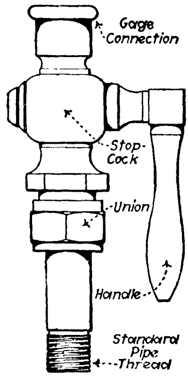


FIG. 250.—Stop cock for steam gage connection.

Pressure on the Boiler (A.S.M.E. Code, par. 297).—This insures that the true pressure will be indicated in the event that the safety valve sticks or if for any reason the steam pressure exceeds that allowable. The gage will also hold its calibration better when selected for this higher pressure.

337. Each Boiler Should Be Provided with a ¼-in. Pipe-size Valved Connection for Attaching a Test Gage (A.S.M.E. Code, par. 298).—This special test-gage connection is for use when checking the accuracy of the boiler steam gage with a standard test gage.

338. A water column is a vessel extraneous to the boiler and suitably connected with it to which water gages and

low- and high-water alarms may be attached. The steam gage may also be attached to the water column. Water columns are on all modern power boilers.

339. The pipe connections for a water column should be at least 1-in. pipe size. It should be fitted with a cross *D* (Fig. 251) at each right-angled turn to permit cleaning. The steam connection to the water column of a horizontal return-

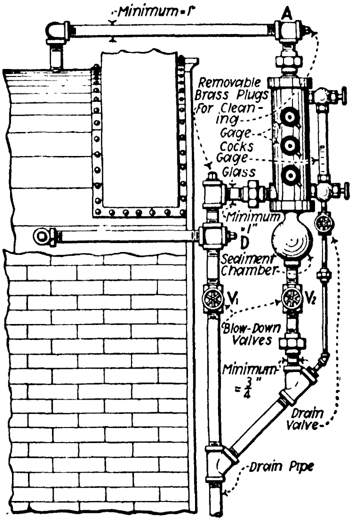


Fig. 251.—A properly-connected water column. (At *A* and *D* either outside-screw-and-yoke-type gate valves or stop cocks, with levers permanently fastened thereto, may be used.)

tubular boiler should be taken from the top of the shell or the upper part of the head. The water connection should be taken from a point not less than 6 in. below the center line of the shell (A.S.M.E. Code, par. P-320 to 322), and must drain toward the boilers throughout its length.

340. The height of the water column relative to the lowest water level that can be maintained with safety is important. When attached to a horizontal return tubular boiler, the water column should be installed at a height that will bring the center of the hole, which is tapped for the lower valve of the glass gage, at least 2 in. above the top row of tubes (Fig. 252).

This insures about 3 in. of water above the tubes when the water is just disappearing from the glass gage. A water column should be of such size and so located that the normal water level is near the center of the column.

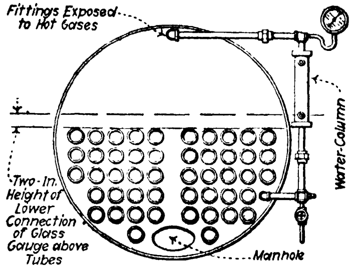


FIG. 252.—Water column set at proper height.

341. Piping between Water Column and the Boiler Should Have No Outlet Connections except for Damper Regulator, Feed-water Regulator, Drains, or steam Cages (A.S.M.E. Code, Par. P-295).—Disregard of this rule (Fig. 253) may lead to trouble. It is important that the full boiler pressure be conveyed to the surface of the water in the column. A flow of steam to some other apparatus ("as to Steam-jets," Fig. 253) than the water column, might result in a decreased pressure above the water in the column. This would cause the water to stand higher in the glass gage than in the boiler. The false indication due thereto might result in an accident. Figure 251, illustrates a correctly connected water column.

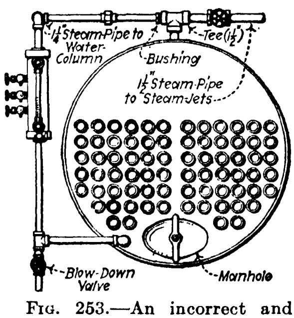


FIG. 253.—An incorrect and dangerous arrangement of water column and steam-jet connected to same outlet from boiler.

342. Shutoff Valves May Be Used in the Connections between the Water Column and the Boiler.—When such valves, are used "they shall be either outside-screw-and-yoke-type gate valves, or stopcocks which have levers permanently fastened thereto, and such valves or cocks shall be locked or sealed *open*." (A.S.M.E. Code, par. 293.)

The locations should be between A and D, Fig. 251, and the boiler. A conservative arrangement of a water-column piping is shown in Fig. 254 wherein the shutoff valves are shown in the horizontal pipes.

343. The glass water gage for moderate pressure (Fig. 255) is usually in the form of a tube which is inserted between valves S and W. These valves control the steam and water passages to the boiler. The level of the water, as seen in the glass tube, coincides with the water level in the boiler. By

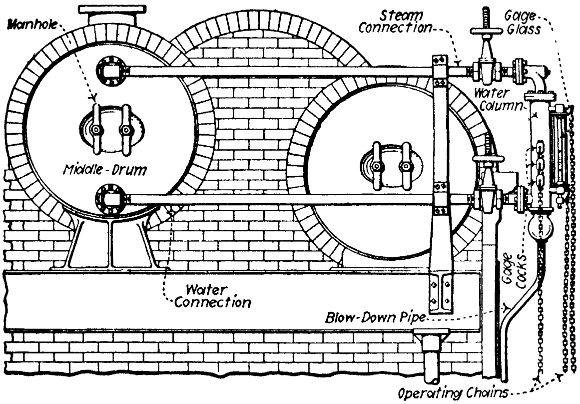


FIG. 254.—Water column and connections on Stirling water-tube boilers.

opening the drain cock D a current of steam is allowed to sweep through the tube and blow out any sedimental deposits.

344. Flat water-gage glasses (Fig. 256) are recommended for pressures over about 400 lb. Steam at higher pressure etches out the tubular-type gage glass which lasts a relatively short time. Two strips of glass rectangular in cross section are bolted between two cover plates and held tightly against a steel body. Sheet-asbestos cushions between steel and glass prevent undue strains on the glass when the covers are tightened down. Thin sheets of mica are inserted between the glass and the steel body to protect the glass from the etching action of the steam.

345. Gage cocks or try cocks (Fig. 251) are supplementary water gages. Each boiler should have at least three of these

cocks. The middle one, when there are three, should be at the normal water level. The higher one and the lower one are placed at the highest and lowest permissible water levels. The lowest cock, when opened, should allow only water to escape, and the highest one, only steam. The intermediate cock may emit a mixture of water and steam if the water is at the correct level. An ordinary wooden-hand-wheel gage cock is shown in Fig. 257. When the gage cocks are so high from the floor as to be difficult to reach, they are operated by chains and are automatic in closing. Figures 257 and 258 show some types of try cocks.

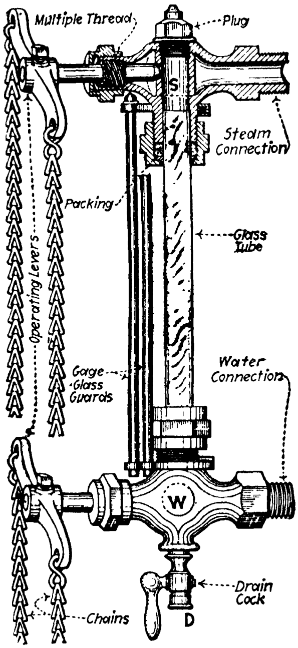


FIG. 255.—Water gage with quick-closing valves.

346. The Water-gage Glass May Be Enclosed.—This provides protection to the operators when a gage glass breaks.

347. To Prevent a Continuous Flow of Water and Steam when a Gage Glass Breaks, an Automatic Valve May Be Used (Fig. 259).—When the gage glass breaks and water and steam begin to pass through the valves the balls take positions over the openings, thus stopping the flow. The balls should be non-ferrous. The ball seat in the upper valve should be square or hexagonal, thus insuring against a tightly seated valve. This is a guarantee that the valve will never become accidently closed, under working conditions. The ball in the lower valve should rise vertically to close the water opening.

348. Low- and High-water Alarms May Be Included in the Water Column.—These alarms consist of an automatically operated whistle which gives a signal when the water becomes too low or too high. Figure 260 shows such an alarm in a

water column. Details of operation may be observed in Fig. 261. High water in the column lifts the upper float U, thus opening the steam passage to the whistle. When the

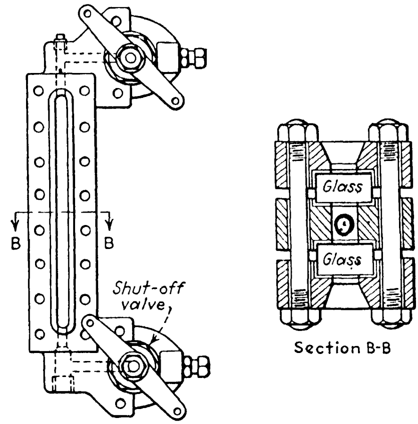


FIG. 256.—Flat-gage glass for high pressures. (*Yarnal Waring Company.*)

water becomes low, the weight of the lower float L opens the whistle valve. Other types may give only the low-water signal.

Figure 262 shows a high- and low-water alarm which depends for its operation upon the displacement action of solid weights

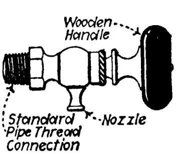


FIG. 257.—Try cock of the simplest form.

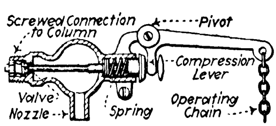


FIG. 258.—Sectional view of Reliance self-closing spring-operating try cock.

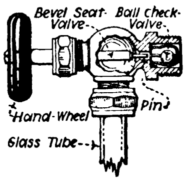


FIG. 259.—The Crosby automatic water-gage valve.

suspended from opposite ends of two levers which pivot on a knife-edge fulcrum supported by lugs on the lower side of water-column over flange. On each lever is a horizontal pin which fits into a yoke on the lower end of the whistle valve stem. The upper end of the stem is fitted with a needle valve which seats on a disk-type valve seat. Needle valve

and seat are made of stainless steel; both are reversible and renewable. The weights are buoyed up by a force equal to the weight of the water they displace. When the water in the boiler is at normal level the upper weight is above the water and the lower weight is submerged. In this position the upper

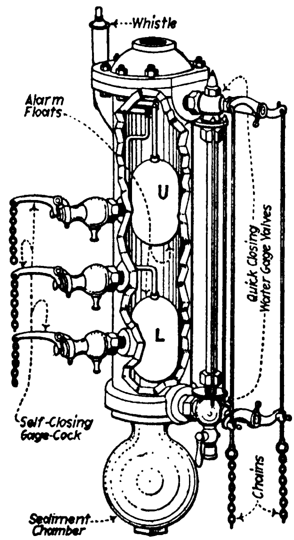


FIG. 260.—Reliance water column showing inside and outside fittings.

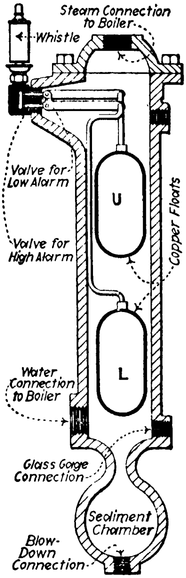


FIG. 261.—Working mechanism of Reliance high- and low-water alarm.

weight is slightly the heavier; the valve is seated, and the alarm is silent. When the water level is lowered to the point at which the lower weight (for low alarm) is emerging from total submergence or when the level is raised to the point at which the upper weight (for high alarm) is becoming submerged, the weights are unbalanced, the lower weight being the heavier. This causes a downward pull which quickly unseats the valve, admitting steam to the alarm. This steam operates the whistle which continues to sound until normal water level has been reestablished.

349. Boiler Blowdown.—Feed water always carries dissolved solids with it into the boiler. These solids are left in the boiler after the water has been evaporated into steam. Consequently, the amount of solids is continuously increasing and unless the concentration is reduced serious trouble will result from foaming, carry over, and possibly burned-out

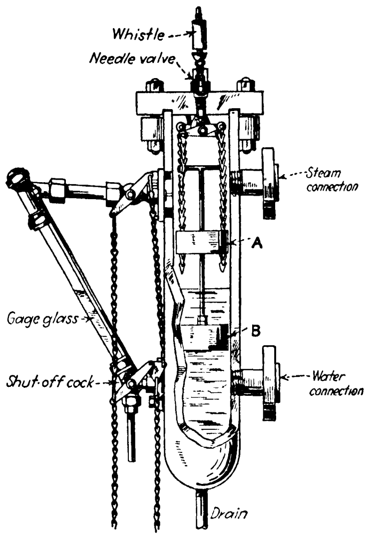


FIG. 262.—Water column with high- and low-water alarms operated by displacement weights. (*Yarnal Waring Company.*)

tubes. The concentration may be reduced by blowing off some of the concentrated boiler water and replacing it with fresh less-concentrated feed water. This can be accomplished by periodically blowing out through blowoff valves a considerable quantity of boiler water or by continuously removing a smaller amount. In either case the amount of blowdown, in per cent of steam output, necessary to maintain a desired concentration in the boiler is found from the following formula:

$$\text{Per cent blowdown} = \frac{\text{concentration of feed water (p.p.m.)}}{\text{concentration of boiler water (p.p.m.)}} \quad (57)$$

Bottom blowoff connections and valves are required by the Boiler Code, but in addition surface blowoff is sometimes also provided for removal of impurities.

350. Requisite number and disposition of blowoff valves of a boiler installation (A.S.M.E. Code, par. P-311) are as follows: Each blowoff connection, when the steam pressure is over 125 lb. per sq. in. gage, should have two slow-opening valves or one slow-opening valve and one quick-opening valve or cock. On boilers having a number of blowoff pipes a single master valve may be placed on the common blowoff pipe from the boiler and only one valve at each individual blowoff connection. The quick-opening valve or cock, if used, should be placed next to the boiler. It should be opened first and closed last to prevent its seats being scored by boiler sludge and so maintain its tightness.

351. Surface blowoff piping should not exceed 1¼-in. pipe size. When inside and outside pipes are used, they should form a continuous passage. This may be accomplished by using bushings or a flange similar to those of Fig. 237. The material for the bushings may be brass or steel (A.S.M.E. Code, par. 307).

352. The bottom blowoff connection should be made to the lowest water-space practicable. Blowoff connection should also be provided for each bottom waterwall header. The size of the pipe should never be less than 1 in. nor greater than 2½ in. A smaller pipe might become clogged by scale. A larger pipe would cause a too rapid loss of water. The piping, when exposed to the products of combustion should be protected by firebrick, a cast-iron sleeve, or a covering of nonconducting material.

353. Since the water does not circulate through the blowoff pipe, it does not carry away the heat. The result is apt to be an overheating which may cause its destruction. Hence the piping should not be exposed to hot furnace gas unless protected. When coverings are provided, they should be such as may be taken off without disturbing the piping. For example, the covering of cast-iron should be split so it may be readily removed and replaced. Figure 263 illustrates a brick protecting pier in front of vertical pipe and a cast-iron

sleeve over the horizontal length. The passage for the pipe through the wall should permit free expansion. The blowoff pipe should be at least extra heavy from the boiler to the valves. No reducers or bushings should be used.

354. The fittings in the blowoff pipe between the boiler and valves (*E* and *T* in Fig. 263) should be made of steel. Cast-iron fittings are not suitable for a blowoff connection. They are liable to crack when subjected to the sudden high-temperature changes incident to the use of the apparatus.

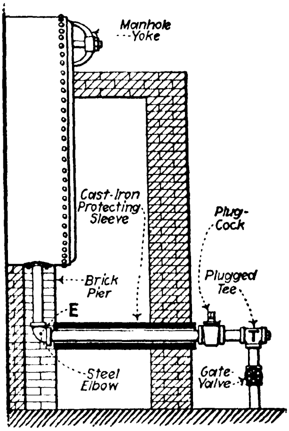


FIG. 263.—Blowoff apparatus of a return-tubular boiler.

355. Special Designed Valves Should Be Used in the Blowoff Piping.—Ordinary globe or gate valves are not suitable. They impede the passage of solid matter in the current of water which issues from the boiler.

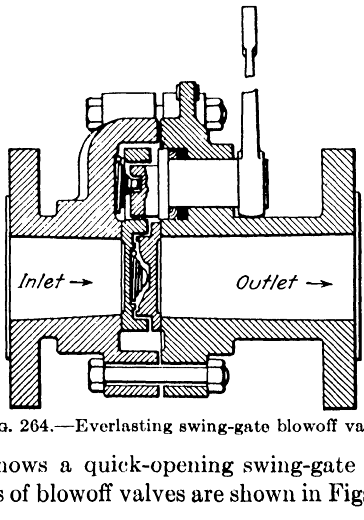


FIG. 264.—Everlasting swing-gate blowoff valve.

Figure 264 shows a quick-opening swing-gate blowoff valve. Typical forms of blowoff valves are shown in Figs. 265 and 266.

356. A blowoff tank (Figs. 267 and 268) is a cylindrical vessel made of boiler plate and set in some convenient location below the level of the boiler blowoff orifice. The boiler blowoff pipe is connected directly to it. Its function is to trap the hot discharge from the boilers. The entrapped water cools in the interval following a blowdown of the boilers. The water thus cooled is displaced by the hot water discharged from the boiler in the next succeeding blowdown. By this means the damage to sewer, which would result from discharging the hot water directly into it, is avoided. The

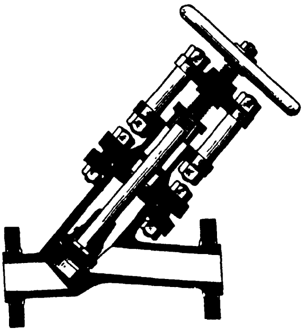


FIG. 265.—Edwards straightway blowoff valve.

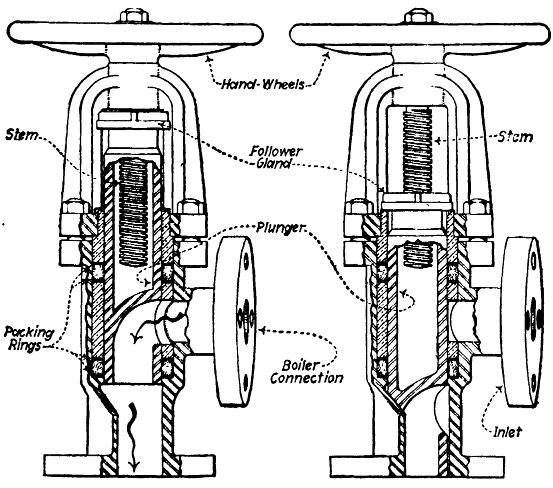


FIG. 266.—Yarway seatless angle blowoff valve.

syphon breaker (Fig. 268) prevents a syphoning action through the outlet extension into the tank.

NOTE.—The piping connections to a blowoff tank should be so proportioned and arranged that the pressure within the tank can not become excessive. Blowoff tanks are not designed as pressure tanks and hence are liable to explosions if subjected to considerable internal pressure. To protect a blowoff tank from explosions, a vent pipe, of a size greater than that of the blowoff-pipe inlet, should be provided direct to the atmosphere. Also the water connection to the sewer should be of a size larger than the steam inlet to the tank.

357. Soot Should Not Be Allowed to Accumulate on the Boiler Heating Surfaces.—Soot is an excellent heat insulating material. Hence the quantity of heat transmitted

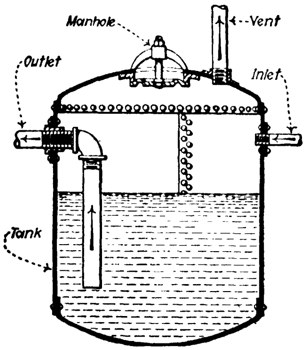


FIG. 267.—A blowoff tank.

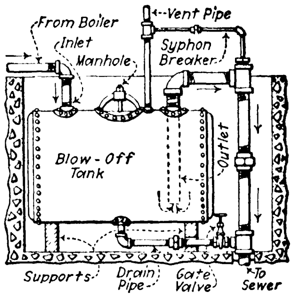


FIG. 268.—Typical blowoff tank with piping pertaining thereto.

to the water in the boiler will be diminished if a coating of soot is permitted to stay on or in the boiler tubes. Soot removals should be frequent—from one to several times per day, according to the fuel and the conditions of firing.

NOTE.—Usually the substance that settles and adheres to the tubes is a mixture of soot, ash, and dust. If this is allowed to remain it may fuse and cake. It thus makes a scale which is difficult to remove. Fresh accumulations of soot on the boiler surfaces may be easily blown off. The word *soot* will be used here to mean a mixture of soot, ash, and dust.

358. Soot may be removed from the boiler heating surfaces in the following ways: (1) by brushing, (2) by scraping, (3) by blowing. If the soot has become caked, the first and second methods must be used. If the accumulation is loose

and flocculent it may be blown off with air or steam blasts. Water spray has been used with success on the first row of tubes to remove ash slag.

Explanation.—Scraping and brushing are especially adapted for removing soot deposits from fire tubes and flues. A typical form of

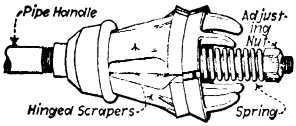


FIG. 269.—Tube scraper.

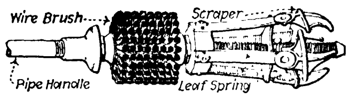


FIG. 270.—Combination brush and scraper.

tube scraper is shown in Fig. 269. When the hardened soot has been cut loose from the tubes with a scraper, it may either be brushed out or blown out with air or steam jets. A combination flue- or tube-brush and scraper is shown in Fig. 270.

359. Boilers Should Be Equipped with Permanently Attached Soot Blowers (Figs. 271 to 274).—Such apparatus mitigate the disagreeable features of soot blowing by hand. Thus they conduce to vigilance in this detail of operation.

360. Each element consists of a tube closed at one end (Fig. 271), extending across the boiler and in which venturi-

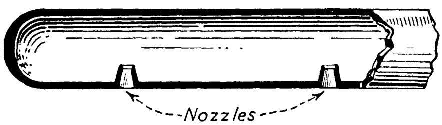


FIG. 271.—Soot-blower element of extra-heavy wrought seamless steel tubing.

shaped nozzles have been welded at intervals corresponding to the boiler tube spacing. This soot-blower element is held by bearings that are clamped or welded to the boiler tubes and permit the element to be rotated. Steam or air is admitted through a soot-blower head, and the jets issuing from the nozzles at high velocity clean off the soot from the tubes. A chain over a sheave wheel permits the operator to rotate the element so that the steam jets will reach all of the tubes in the vicinity of the blower. Soot-blower elements are made of alloy steel or "calorized" (impregnated with aluminum) so as to be capable of withstanding the high temperature of the flue gas.

361. Valve in head soot blowers (Figs. 272 and 273) are made so that rotation of the element automatically opens the steam valve at the proper position of the nozzles and closes the valve when the arc of soot blowing is completed. The valve in the head of Fig. 272 is operated by a cam and closed by a spring. Note also in this head the swivel joint between the head and the soot-blower element which permits some misalignment. In Fig. 273 the cam operates a pilot valve which is normally held open by steam pressure, and in this position it communicates pressure to the back of the main valve thus holding it closed. When the cam closes the pilot, steam pressure back of the main valve is released and pressure on the front side of the valve opens it. The pilot is held in the closed position by a latch until the soot blower is rotated through the blowing arc when a trip releases the pilot. The pilot opens allowing steam pressure on the back of the main valve which is then closed by the spring.

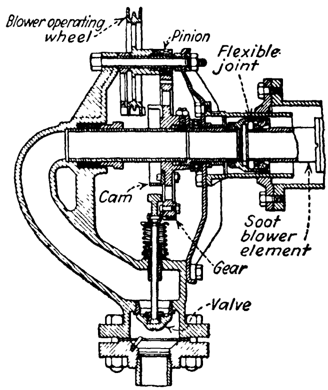


FIG. 272.—Cam-operated valve-in-head soot blower. (*Diamond Specialty Company.*)

362. Soot blowing should start with the element nearest the furnace and progress along the gas flow to the element nearest the boiler outlet. With this method of operation the soot is blown progressively through the boiler and finally out to the stack.

363. Dry Steam or Dry Air Should Be Used in the Operation of Soot Blowers.—Wet steam or air causes some of the soot to stick and cake. The caking of soot, from this cause, in water-tube boilers may progress to the extent of filling up the spaces between contiguous tubes. Such deposits can be loosened and removed only by jabbing them with iron rods or pokers.

364. Proper soot blowing improves the economy of boiler operation from 2 to 10 per cent. This has been demonstrated

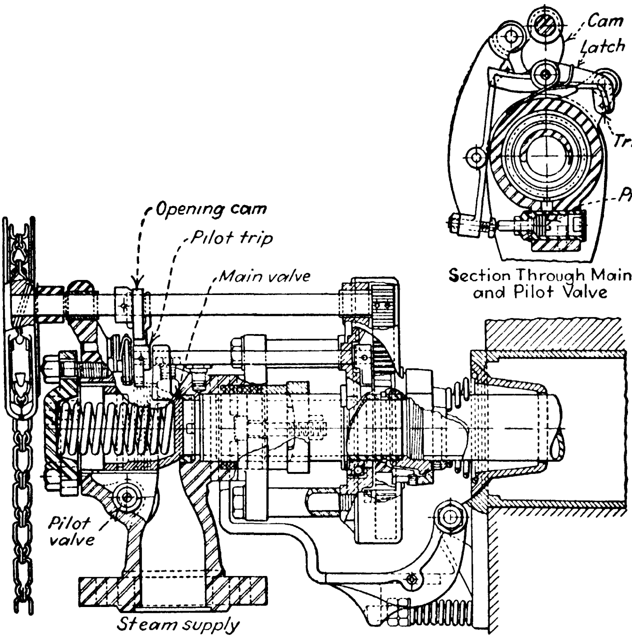


FIG. 273.—Soot blower head operated by pilot valve. (*Vulcan Soot Blower Corp.*)

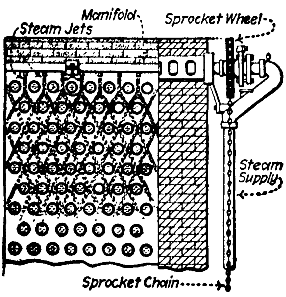


FIG. 274.—Permanently attached soot blower in a water tube boiler. (*Diamond Power Specialty Co.*)

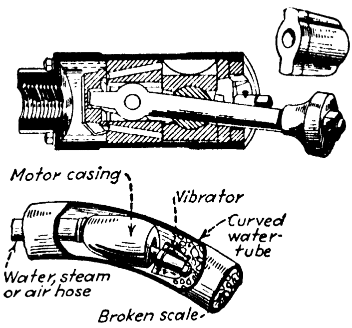


FIG. 275.—Vibrating hammer-head tube-cleaner for curved tubes. (*William B. Pierce Company.*)

by tests. The rise in temperature of the flue gases after a day's run without blowing off of soot may be as much as 75°F. After several days' omission, the rise may be near to 175°F.

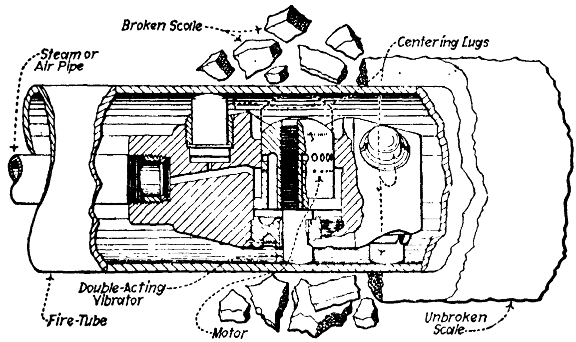


FIG. 276.—Vibrating tube-cleaner.

NOTE.—The cause of scale on the water surfaces of boilers, and the methods of scale-prevention and removal by the action of substances introduced into boilers with the feed-water, are discussed in Div. 24.

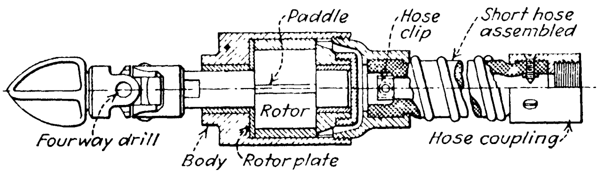


FIG. 277.—Lagonda rotary-tube cleaner.

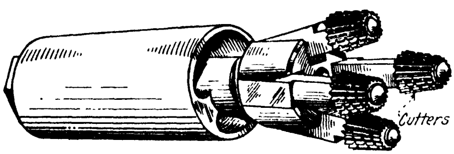


FIG. 278.—Bulldog cleaner for 4-in. straight water tubes. (*Liberty.*)

365. Mechanical devices for removing scale from the water surfaces of boiler tubes are of two general types, as follow: (1) those which jar the scale loose by the hammering action of a vibrating tool (Figs. 275 and 276); (2) those which cut the scale loose by the cutting action of a revolving tool (Figs. 277 and 278). Cleaners of the first type may be used in

water tubes (Fig. 275) or in fire tubes (Fig. 276). Those of the second type are suitable only for water tubes. Those of either type are adapted for use in both straight and curved tubes.

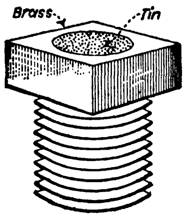


FIG. 279.—A fusible plug.

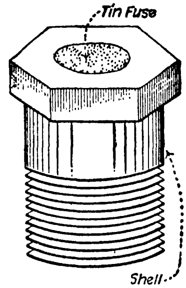


FIG. 280. Fusible plug for inside insertion.

366. The energy for operating mechanical scale removers may, for water-tube cleaners, be transmitted to the encased motors through the medium of steam, air, or water pressure. For fire-tube cleaners the medium may be either steam or air.

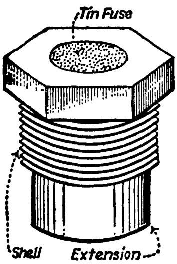


FIG. 281.—Fusible plugs for outside insertion.

NOTE.—The operating motors of mechanical scale removers are miniature engines which embody various applications of the turbine or rotary principle. See author's "Steam Engines."

367. A fusible or safety plug (Figs. 279 to 281) is a brass plug having a core made of some metal which melts at a comparatively low temperature. Its function is to safeguard the boiler against damage from low water.

368. Fusible plugs for direct attachment to boilers are of two general types, as follows: (1) those which are inserted (Fig. 282–*I*) from the fire side of the boiler metal; (2) those which are inserted (Fig. 282–*II*) from the water side. Either type of plug is attached at some point coincident with the lowest level to which the water may recede with safety.

Normally the inner end is covered with water. The outer end is exposed directly to the heat of combustion. When the water level falls low enough to uncover the plug, the heat which accumulates in the core causes it to fuse and blow out. Warning of the danger is thus given. The fire may be quenched at the same time (Fig. 438).

NOTE.—According to A.S.M.E. Code, the plug shall be filled with tin which has a melting point between 400 and 500°F. The tin shall be renewed once each year. The least diameter (Fig. 282) of the fusible metal in the plug shall not be less than 0.5 in. when the boiler pressure is less than 175 lb. per sq. in. If the pressure is greater than 175 lb. per sq. in. or if the plug is for insertion in a tube the least diameter of the fusible metal may be 0.375 in. The location of the fusible plug in a horizontal return-tubular boiler shall be in the rear head, not less than 2 in. above the upper row of tubes as measured from the line of the upper surface of the tubes to the center of the plug. The plug shall project not less than 1 in. (Fig. 282) inside the boiler plate. The locations for fusible plug in boilers of various types are given in A.S.M.E. Code, par. A-21.

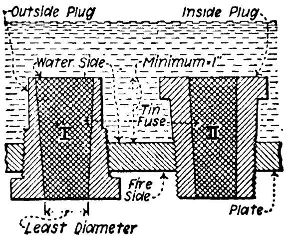


FIG. 282.—Fusible plugs inserted in boiler plate.

369. A feed-water regulator (Figs. 283, 284 and 285) is a device which automatically controls the flow of feed water to a boiler so as to maintain proper water level.

370. Feed-water regulators are of two principal types, as follows: (1) those which maintain a constant water level in the boiler; (2) those which allow a considerable variation in the quantity of water contained in the boiler. With the latter type of regulator, the water level is permitted to fall to a definite extent when sudden heavy loads are thrown on the boiler and is permitted to rise to a definite extent during intervening light-load periods.

371. The advantages claimed for automatic feed-water regulation are principally as follows: (1) Accidents on account of either high or low water are avoided. (2) Local stresses, on account of contractions due to intermittent injections of large quantities of cool water, are avoided. (3) Loss of effectiveness of feed-water heating apparatus, on account

of intermittent rushes of cool water, is avoided. (4) Loss of furnace economy, on account of the irregular firing which an intermittent feed makes necessary is avoided. (5) The ability of the boiler to deliver dry stream is fully realized.

372. Water-storage space in boiler drums has decreased considerably compared to boiler output making the duty of automatic feed-water regulators more exacting. No one type of regulator can meet all the widely varying demands

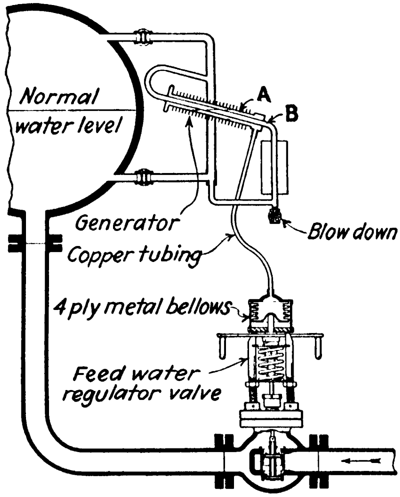


FIG. 283.—Thermo-hydraulic system of feed-water regulation. (*Bailey Meter Company.*)

of modern boiler practice, some boilers requiring a regulator feeding according to the rate of steam flow, others being best handled by regulation from water level. Figure 283 shows a feed-water system controlled by water level and Fig. 285 a system controlled by steam flow and water level. To meet exacting demands of some installations, a pressure regulator is sometimes placed ahead of the feed-water-level control valve to maintain a constant excess pressure ahead of the control valve.

Explanation.—The thermo-hydraulic control of Fig. 283 depends for its operation on the pressure developed in the generator which is transmitted

to the inside of the metal bellows on a balanced control valve. The force exerted by the bellows, due to the pressure, is opposed by a spring which holds the valve closed. The generator consists of two tubes, the outer one having cooling fins and being sealed around the inner tube which is connected to the boiler drum with one end above and one below the water level. The generator is inclined and placed so that normal water level comes at its center. With normal level the inner tube is half full of water and half full of steam. A closed system is formed by the annular space in the generator, the valve bellows, and the connecting copper pipe which is filled with water before the control is put into operation. When the regulator is placed in operation, heat from the steam in the upper portion of the inner tube causes the surrounding water to turn into steam thus forcing water out of the annular space until its water level is the same as that in the inner tube and hence the same as the level in the boiler drum. Water forced out of the annular space expands the metal bellows and opens the regulating valve. If the boiler water level tends to rise, cold water from the water-storage leg rises into the inner generator tube. This cold water plus radiation from the outer tube causes steam in the annular space to condense, thereby reducing the generator pressure, and the regulating valve spring forces water back to the generator leg compressing the bellows and at the same time partially closing the regulating valve. The control is so proportioned that the valve is wide open when the annular space is filled with steam and closed when it is filled with water.

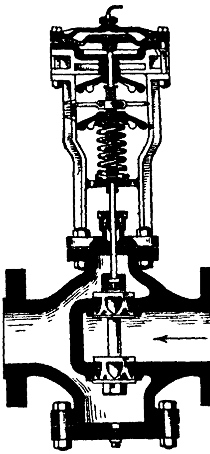


FIG. 284.—Feed-water regulating valve such as used in a system similar to that of Fig. 283. (*Swartwout Company.*)

The feed-water control system of Fig. 285 has two thermostatic elements, one influenced by steam flow the other by water level. Each element consists of a tube having a high coefficient of expansion that is held between two channels. Its lower end is fixed and the upper end is connected to a bell crank. A weight on the control-valve lever arm to which the thermostats are connected as shown keeps the tubes in tension. The lower end of the tube in the level thermostat is connected to the water space of the drum and its upper end is connected to the steam space. The thermostat is placed at an incline so that its center is at the same elevation as normal water level in the boiler drum. The steam-flow thermostat is placed so that the top of the tube is level with the water in the reservoir connected to the downstream pressure tap of an orifice in the steam discharge pipe. The upstream orifice pressure tap is connected to the top of the expansion tube; downstream tap is connected

to the lower end of the tube. Steam flow creates a pressure drop across the orifice which causes the water level in the tube to drop as the flow increases. Steam is hotter than the water and hence the tube expands allowing the feed valve to open. Similarly the water-level thermostat expands if the level in the boiler falls thus allowing the feed valve to open further. The action of the steam-flow thermostat is to partially offset the tendency of the water-level thermostat to permit lower levels

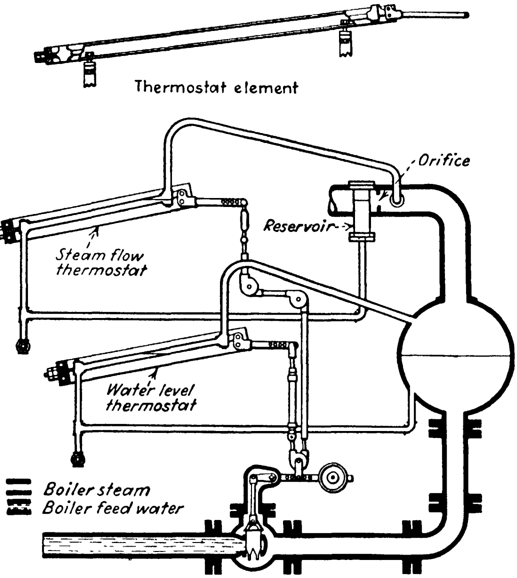


FIG. 285.—Feed water controlled by steam flow and water level. (*Copes.*)

at high loads and high level at low load. In the majority of installations only the water-level thermostat is used.

373. A steam-flow meter is an instrument for measuring the quantity of steam flowing through a pipe. This indicating instrument simply indicates the quantity of steam, in pounds per hour, flowing at any instant. The recording steam-flow meters draw a continuous graphic record indicating the quantity of steam, in pounds per hour, which flows at any instant during an extended interval of time. The simple indicating instrument is often used as a boiler accessory. When a steam-flow meter is connected to the steam outlet

of each boiler in an installation, the work of evaporation performed by each boiler, when all of them are working in unison, may be definitely known.

NOTE.—For a discussion of steam-flow meters, see the author's "Practical Boiler Room Economy."

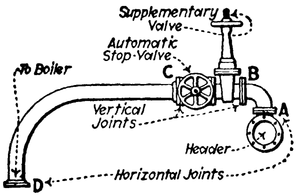


FIG. 286.—An approved form of steam-outlet piping for a boiler.

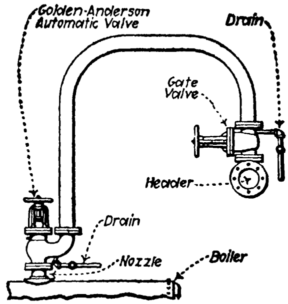


FIG. 287.—Boiler steam-outlet pipe in form of U-bend.

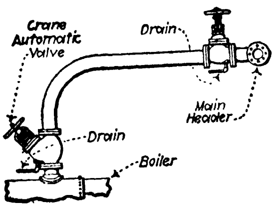


FIG. 288.—Boiler steam-outlet piping with rigid header connection.

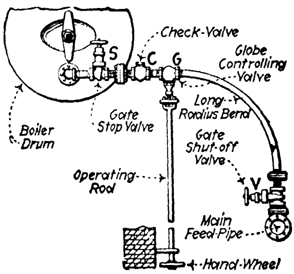


FIG. 289.—Branch feed pipe.

374. The branch live-steam piping which connects a boiler with the main header should be erected with a view to avoiding excessive stresses during erection and those due to subsequent expansion and contraction when the pipe is heated by the steam.

375. A drain pipe should be connected to the body of the automatic non-return valve (Figs. 287 and 288) in every case where the piping is such that water resulting from condensa-

tion of steam coming from the main header will collect above the valve disk when the latter is firmly seated.

376. The arrangement of the feed piping (Fig. 289) should be such as to afford convenience in regulating the feed and assurance against trouble on account of obstructions lodging in the pipe.

377. The feed-piping fittings, such as elbows, tees, and flanged unions, should be extra heavy. Water hammer from some unforseen cause might develop in the piping while the boiler is in operation and a cast-iron fitting might break under the strain. In the general design of a feed-water piping system, long-radius bends should be used, whenever possible, instead of short right-angled fittings.

NOTE.—See A.S.M.E. Code, par. 317 for feed-pipe valve requirements. In Fig. 289, *S*, *C*, and *G* are required by the Code. *V* is installed so that *C* and *G* may be repaired or reground without shutting down.

QUESTIONS ON DIVISION 14

1. What are boiler accessories?

2. What do you regard as the least number of accessory appliances with which a boiler can be safely operated? Name them.

3. Upon what general principle of construction does the prompt action of spring-loaded safety valves depend?

4. Are weighted safety valves permitted?

5. A spring-loaded safety valve opens when the pressure in the boiler is 150 lb. per sq. in. and continues to blow until the pressure is 130 lb. What should ordinarily be done to cause it to close at, say, 142 lb.?

6. What metals should be used in the disks and seats of safety valves? Why will not iron or steel suffice for this purpose?

7. What may cause chattering of safety valves?

8. What precautions should be observed in the piping of safety valves?

9. What is an accumulation test?

10. What precautions should be observed as preliminary to an accumulation test?

11. What considerations govern in determining whether one or two safety valves shall be used on a boiler?

12. What factor determines the requisite relieving capacity of a safety valve? What is the value of this factor for a water-tube boiler? For any other type of boiler?

13. What factors mainly govern in making a choice of location for the feed-water inlet to a boiler?

14. How should the feed-water connection to a horizontal return-tubular boiler be arranged?

15. What might be considered an inhibitive defect of the spray method of feeding a boiler?

16. What advantage would a spray-feed secure? Would not this advantage be multiplied by the disadvantage of the device?

17. What is a check valve? What purpose does it serve in a feed line?

18. What good purposes are served by automatic stop valves in the steam-outlet connections of boilers?

19. How may automatic non-return stop valves conserve the economy of a boiler plant?

20. What are the advantages of installing an ordinary stop valve in conjunction with an automatic stop valve in the steam-outlet connection of a boiler?

21. What is the basic pressure in the reading of an ordinary steam gage?

22. What is the operating principle of an ordinary steam gage?

23. What harm would result from admitting steam directly to a steam gage? How may this be prevented?

24. What is a water column?

25. In the installation of a water column, how should its proper height be gaged?

26. What should be the minimum size of the steam and water connections of a water column in any case?

27. What are the restrictions of the A.S.M.E. Code in regard to valves in water-column connections?

28. What is the objection to tight-closing automatic valves in the glass water-gage connections?

29. What should be the height, above the tubes, of the lower valve of the glass water gage on a horizontal return-tubular boiler? What should be the location of water column on a water-tube boiler?

30. How do high- and low-water alarms operate?

31. What benefit results from periodic blowing off of the surface water in a boiler?

32. What should be the maximum size of the surface blowoff piping in any case?

33. What benefit results from periodic blowing down of the water in a boiler?

34. Why is it necessary to shield the bottom blowoff pipe from the direct action of the fire?

35. What are the minimum and maximum pipe sizes permissible for bottom blowoff connections? Why are these restrictions necessary?

36. What materials should be used in the boiler blowoff fittings?

37. What are the requirements of the A.S.M.E. Code regarding blowoff valves?

38. Why are ordinary globe and gate valves ill-adapted for blowoff service?

39. What is the purpose of a blowoff tank?

40. Of what are sooty deposits in and on boiler tubes composed?

41. How may soot be removed from fire tubes? From water tubes?

42. What are the advantages of permanently attached soot blowers?

43. What percentage of saving may be realized by systematic soot blowing in a boiler plant?

44. What are the operating principles of mechanical scale removers?

45. What is the object of inserting fusible plugs in boilers?

46. What are the requirements of the A.S.M.E. Code regarding the location of fusible plugs in horizontal return-tubular boilers?

47. What is the standard fusible metal for use in fusible plugs?

48. What are the two principal types of feed-water regulators?

49. What advantages are claimed for automatic feed-water regulation?

50. What is a steam-flow meter?

51. What is the advantage of fitting each boiler in an installation with a steam-flow meter?

52. What should be the foremost consideration when planning the branch piping from a boiler to the main steam header?

53. Why should automatic non-return stop valves be fitted with drain pipes?

DIVISION 15

FUELS

378. The Principal Steam Fuels Are Coal, Oil, and Gas. Of these, coal is the most common. The fuels herein enumerated and others are discussed more fully in the author's "Practical Heat."

379. The calorific values of the principal steam fuels, in B.t.u. per pound of air-dry fuel, are approximately as follows: (1) anthracite, from 12,800 to 14,000, (2) semibituminous, from 14,400 to 12,000; (3) bituminous, from 10,000 to 14,500; (4) subbituminous, 8,500 to 11,000; (5) lignite, from 6,500 to 9,500; (6) oil, from 18,000 to 21,000; (7) national gas, from 1,000 to 1,200. See the author's "Practical Heat" for more complete information.

NOTE.—A fuel is "air-dry" after it has been spread out in a thin layer and exposed to the air of a warm room for several hours.

NOTE.—Tests to determine the relative calorific values of fuel oil and coal indicate that 1 lb. of oil is the equivalent of about 1.37 lb. of bituminous or 1.67 lb. of anthracite coal. A maximum evaporation of about 15 to 16 lb. of water per pound of fuel has been obtained with oil as against about 11.5 to 12 lb. of water per pound of combustible with coal.

380. Anthracite is commonly called *hard coal.* It is very dense. Its specific gravity is high. It has a metallic luster. It burns with a short yellowish-blue flame and without smoke. It is largely carbon. Analyses of various grades (air-dry) show an average of about 83.5 per cent fixed carbon 3.2 per cent volatile matter, 10.5 per cent ash, and 2.8 per cent moisture.

381. Commercial sizes of anthracite are, from the smaller to the larger sizes, called *buckwheat, pea, chestnut, stove egg,* and *broken.* Only the smaller sizes are used in power boilers. Sizes are determined by square mesh screens and are as shown in Table XI.

382. Semianthracite is less hard than anthracite. It is less dense, ignites more readily, and is of somewhat duller metallic luster. An analysis (air-dry) may show about 80.0 per cent fixed carbon, 9.4 per cent volatile matter, 9.3 per cent ash, and 1.3 per cent moisture.

TABLE XI.—COMMERCIAL SIZES OF ANTHRACITE

Commercial or trade name	Size of standard square mesh, in in.	
	Through	Over
Broken	4	2¾
Egg	2¾	2
Stove	2	1⅜
Chestnut	1⅜	¾
Pea	¾	½
Buckwheat:		
No. 1	½	¼
No. 2	¼	⅛
No. 3	⅛	1/16

NOTE.—Boiler-fuel anthracite is generally one of the last three sizes.

383. Semibituminous coal resembles anthracite more closely than it does bituminous coal. It is lighter than anthracite and ignites more readily. It is a very desirable steam coal. Its combustion produces intense heat and very little smoke. It leaves little or no clinker. Analysis (air-dry) of an average grade may show 73 per cent fixed carbon, 17 per cent volatile matter, 2 per cent moisture, and 8 per cent ash.

384. Bituminous or soft coal appears in a wide diversity of grades and sizes. It is difficult to form precise distinctions between the different grades. The color ranges from very black to brown. The more dense varieties show a resinous luster. Those less dense show a silky luster. Bituminous coals generally burn with a yellow flame and much smoke.

385. Coking and noncoking classification of bituminous coals is an important division as this quality largely influences the selection of fuel-burning equipment. A coking or caking coal swells and softens so that particles of the coal stick together forming crustlike masses that must be broken up

before air for combustion can pass through. Coking coals are usually high volatile. A noncoking or free-burning coal does not soften and become sticky and hence does not form coke masses. With such a coal the fuel bed does not require agitation or tempering.

386. Cannel coal is a noncoking bituminous coal. It kindles easily and burns freely with a bright flame similar to a candle flame. From this circumstance it derives its name. It contains about 50 per cent volatile matter. This renders it valuable for gas making. Its structure is very compact. It has a dull luster. When a lump of cannel coal is broken, the fracture will appear to occur along no definite cleavage line.

387. Subbituminous coal is a lower grade of fuel than bituminous coal. Its fuel value is intermediate to those of the true bituminous coals and the lignites.

388. Lignite is apparently of more recent fossil origin than the various coals. It ranges in color from brown to black. It is of woody texture. It is very soft. When exposed to the weather it readily absorbs moisture, and disintegrates by crumbling. Generally, it must be burned in the locality where mined. This is due to its tendency to break up in transportation. Dry lignite may contain over 40 per cent volatile matter and 7 per cent ash.

NOTE.—Newly mined lignite may contain 35 per cent of moisture. Some anthracites and bituminous coals contain as low as 2 per cent of moisture.

389. The sizes and grades of the different bituminous coals are not universally standardized. The coal fields of the United States are approximately separated into two grand divisions—the Eastern and the Western. In each of these a separate size and grade schedule is recognized.

NOTE.—The following classification of bituminous coals according to size and grade is adapted from Mark's "Mechanical Engineer's Handbook."

1. *Eastern Bituminous Coals*.—(*a*) *Run of mine coal* is the unscreened coal as taken from the mine. (*b*) *Lump coal* is that which passes over a bar screen with openings 1¼-in. wide. (*c*) *Nut coal* is that which passes through a bar screen with 1¼-in. openings, and over one with ¾-in. openings. (*d*) *Slack coal* is that which passes through a bar screen with ¾-in. openings.

2. *Western Bituminous Coals.*—(*a*) *Run of mine coal* is the unscreened coal as taken from the mine. (*b*) *Lump coal* is divided into 6-in., 3-in., and 1¼-in. *lump*. These sizes are determined by the diameters of the circular openings over which the respective grades will pass. There is also 6 × 3-in. *lump* and 3 × 1¼-in. *lump*. These sizes depend on passage of the coal through a circular opening whose diameter is expressed by the larger figure and over one whose diameter is expressed by the smaller figure. (*c*) *Nut coal* is divided into 3-in. *steam nut*, which passes through an opening 3 in. in diameter; 1¼-in. *nut*, which passes through a 1¼-in. diameter opening and over a ¾-in. diameter opening; ¾-in. *nut*, which passes through a ¾-in. diameter opening and over a ⅝-in. diameter opening. (*d*) *Slack coal* is that which passes through a ⅝-in. diameter opening. (*e*) *Washed sizes* are those which pass through or over circular openings as given in Table XII.

TABLE XII.—WASHED SIZES OF WESTERN BITUMINOUS COAL

Size number	1	2	3	4	5
Diameter hole, in.:					
Through	3	1¾	1⅛	¾	¼
Over	1¾	1⅛	¾	¼	0

390. Fuel oil (see also Div. 18) is a mineral oil. It is composed of a series of hydrocarbons in various proportions. Crude petroleums from different fields show different compositions. In general, the oil is composed of from about 84 to 85 per cent carbon: 11.5 to 14.5 per cent hydrogen, and a small percentage of other substances. Oil fuels are discussed more fully in the author's "Practical Heat."

391. Gases, natural and artificial, are treated in the author's "Practical Heat."

QUESTIONS ON DIVISION 15

1. State the average calorific values for the principal steam fuels.

2. What is the relative calorific value of coal as compared with oil?

3. What is anthracite?

4. What sizes of anthracite are most commonly used for steaming purposes?

5. What are the characteristics of semianthracite? Bituminous coals? Semibituminous coals? Subbituminous coals? Lignite?

6. Explain what is meant by (1) coking coal, (2) noncoking coal, (3) cannel coal.

7. Discuss the sizes and grades of bituminous coal.

8. What is the general composition of petroleum?

DIVISION 16

COMBUSTION AND HAND FIRING

392. The combustible in a steam fuel comprises those portions which undergo combustion or burning. It is commonly regarded as the sum total of the constituents which remain after deduction of the moisture and ash. These "combustible" constituents, generally, are carbon, hydrogen, and sulphur.

393. Combustion may be defined as the chemical union, at a rate sufficiently rapid to produce a high temperature, of oxygen with the combustion in a fuel. This subject is treated fully in the author's "Practical Heat."

394. The oxygen necessary for combustion is obtained from the air. Air is composed, by weight, of about 23 per cent oxygen and 77 per cent nitrogen. By volume, the proportions are 21 per cent oxygen and 79 per cent nitrogen. Nitrogen is an inert gas. Hence it does not combine with other substances to produce combustion. Small quantities of other gases are present in air. These, however, do not affect combustion.

395. The process of combustion in a boiler furnace is very complicated. But regardless of what chemical reactions occur, the combustion should, in general, be complete to insure maximum economy. The products of complete combustion are (1) carbon dioxide gas, CO_2; (2) water, H_2O; (3) nitrogen gas, N_2; (4) a small quantity of sulphurous acid gas, SO_2. If combustion is incomplete, the products may contain some carbon monoxide gas (CO). There will then be a correspondingly smaller quantity of CO_2. Hence the completeness of combustion may be gaged quite accurately from the amounts of CO and CO_2 in the discharge gases.

NOTE.—Usually, 10 to 15 per cent, by volume of CO_2 indicates efficient combustion. In practical boiler operation, by using apparatus

which indicates the CO_2 content in the discharge gases, the completeness of the combustion may be judged. Furnace economy can then be controlled accordingly. (See author's "Practical Heat.")

396. The quantity of air actually required for the combustion of coal is commonly considered to be about 12 lb. for each pound of coal.

NOTE.—It has been determined ("Finding and Stopping Waste in Modern Boiler Rooms," Cochrane Corp.) that the quantity of air requisite for the combustion of 1 lb. of coal may vary from 7 to 11.2 lb. Hence it is suggested that the air supply to a boiler furnace be expressed as pounds of air per 10,000 B.t.u. On this basis, combustion of coal requires about 7.5 lb. of air per 10,000 B.t.u. generated.

397. The Quantity of Air Actually Required for Combustion Is Always in Excess of the Quantity Theoretically Required.—This is due to the impracticability of securing, if no more than the theoretically requisite quantity of air were admitted to the furnace, a perfectly adequate mixture of the oxygen with the combustible. Usually, from 30 to 50 per cent excess air should suffice to insure complete combustion. An excess of 200 per cent and even 400 per cent is not uncommon. This results in great loss, since these inordinate volumes of excess air act only to carry heat from the furnace.

398. The requisites for proper combustion. ("Principles of Combustion in the Steam Boiler Furnace," A. D. Pratt) are:

1. The admission of an air supply such as will assure sufficient oxygen for complete combustion.
2. Since complete combustion is not, of necessity, efficient combustion, it must be secured without permitting dilution of the products of combustion with excess air. It follows then, that:
3. The air supply should be admitted at the proper time and in such a manner that the oxygen of the air will come into free and intimate contact with the combustible substances in the fuel. In the case of solid fuels this means not only into contact with the solid particles of the oxidizable substances, but also with the combustible gases as they are distilled from the fuel.
4. The gases must be maintained at a temperature at or above their ignition point until combustion is complete. Theoretically, the most efficient combustion is that resulting in the maximum temperature possible. In practice, there are, frequently, factors which, from the standpoint of commercial operating efficiency, make it advisable to

keep furnace temperatures somewhat below those which could be obtained were this the sole factor involved.

5. An additional requirement, which has to do with the physical rather than the chemical aspect of combustion, is that proper provision must be made for the expansion of the gases during the period of their combustion.

399. The Efficiency of Combustion Is Determined Solely from the Chemical Changes That Occur in the Burning Com-

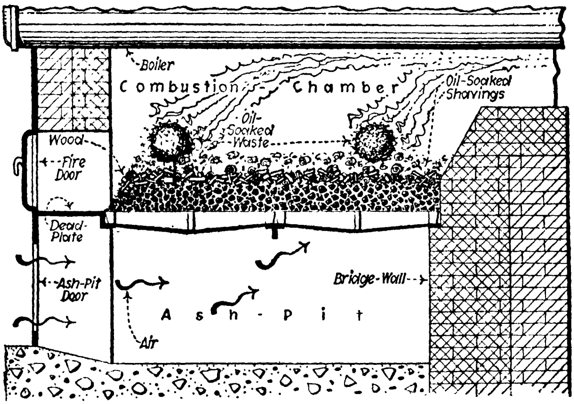


FIG. 290.—Starting a fire with soft coal in hand-fired furnace.

bustible.—It is independent of all considerations with respect to the ability of the boiler to absorb the heat which is generated.

NOTE.—Certain physical and mechanical factors, involving details both of furnace manipulation and structure, render difficult the attainment of proper combustion. With furnaces of ideal construction, and with adequate combustion temperatures, the problem would be solely one of air admission and admixture. It would be necessary, however, to adapt the methods of furnace operation in accordance with the peculiarities of various fuels.

400. Anthracite when hand-fired should be spread evenly, in small charges, at frequent intervals. Stirring or breaking up the fuel bed with the slice bar or poker should not be practiced. The thickness of the fires is, usually, not over 2 or 3 in., but the fuel bed builds up between cleaning intervals. The total thickness may, therefore, be from 14 to 16 in. just before cleaning. The cleaning periods should be regulated in accordance with the combustion rate and the quantity of

the fuel. When cleaning fires, it should be remembered that anthracite is slow to ignite and that a sufficient quantity of glowing fuel should, therefore, be retained.

NOTE.—Artificial draft is usually employed in burning anthracite. For burning the smaller sizes, a blast equivalent to a 3-in. water column should be available. The stack should be so proportioned as to give a suction at all times within all parts of the boiler setting. A forced blast with anthracite causes rapid fouling of the heating surfaces. The dust carried over often amounts to 10 per cent of the total quantity of coal fired. (Marks', "Mechanical Engineer's Handbook.")

401. To start a fire with soft (bituminous) coal the following method (Maujer and Bromley, "Fuel Economy in Boiler

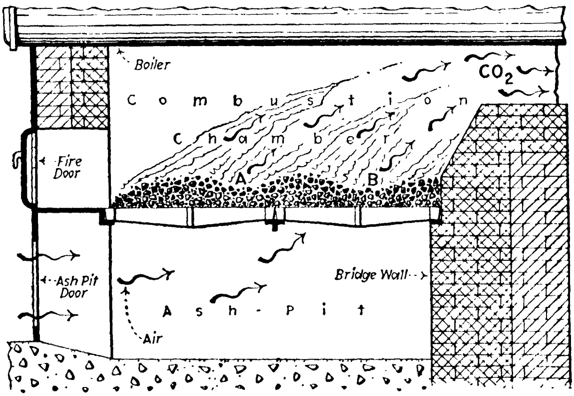


FIG. 291.—A well kept fuel bed before firing.

Rooms") may be followed: The entire grate is covered (Fig. 290) with 3 in. of green coal spread evenly. Dry wood or shavings are then spread on top of the coal. On these, here and there over the surface, is put oil-soaked shavings or waste. The fuel is then ignited by throwing burning waste in the center of the grate. The blower is started lightly at first, or the damper and ash-pit doors are opened if no blower is used. The draft may be increased as necessary, coal should be thrown on, a little at a time, until the fire is going satisfactorily. If a lumpy gas coal is available it should be first spread on the grate. The coal ordinarily used and the shavings may then be laid. This will prevent the small coal

falling through the grate. The fire burns from the top down. The volatile passes through the hot zone and becomes well vaporized, so that it mixes readily with the air supplied for combustion.

402. Hand firing of bituminous coal ("Hand Firing Soft Coal under Power-plant Boilers," Henry Kreisinger, Government Printing Office) secures good results, generally, only

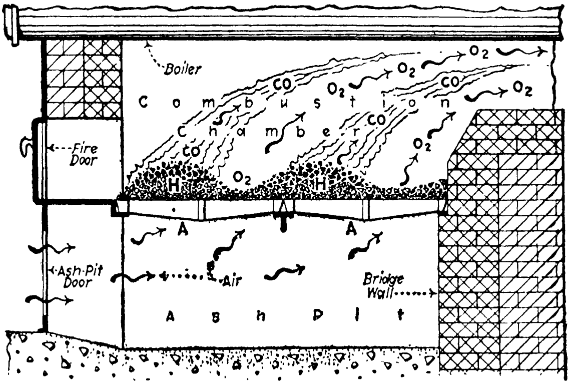


FIG. 292.—Condition after several firings when fuel is spread evenly over whole grate without noting thick and thin spots.

when certain fundamental rules are scrupulously observed. These are:

(1) The fire should be kept level by depositing the green fuel only where the fire (*A* and *B*, Fig. 291) shows signs of thinning out. Deep depressions in the fuel bed (Fig. 292) should not be filled at one firing. An excessive depth of green coal might cake and retard combustion in the depression. At the same time, replenishment of the high places, *HH*, should be omitted for one or two firings.

(2) A proper thickness of fuel bed should be maintained. This factor is contingent, mainly, upon the grade and quality of the coal and the strength of the draft. It may be from 4 to 10 in.

(3) The coal should be fired in small quantities and at short intervals. This tends to keep the fuel bed level and prevents the formation of large volumes of dense black smoke.

(4) The fuel bed should not be leveled or otherwise disturbed with the slice bar or poker. If the fire is leveled with the slice bar, the furnace doors must remain open during the operation. A large excess of air will thus be admitted to the furnace. Impaired efficiency will result. Stirring the fire bed will inevitably lead to difficulty if the coal is of a clinkering variety. This will be due to fusing of the ash by mixture with the glowing fuel.

(5) The ashpit doors should be kept open at all times while the boiler is being operated. The draft should be controlled with the damper, not with the ashpit doors.

(6) Excessive accumulations of furnace refuse in the ashpit should be avoided. Clogging of the ashpit in this manner may result in an uneven distribution of the air supply under the grate.

NOTE.—Proper thickness of the fuel bed for different coals (Mark's "Mechanical Engineers' Handbook") are: (1) For semibituminous coals, as Pocahontas, New River, Clearfield, from 12 to 14 in.; (2) for bituminous coals from the Pittsburgh mining district, Ohio, Illinois, Kentucky, and Tennessee, from 4 to 6 in.; (3) for free-burning coal from the Rock Springs, Wyo., mining district, from 6 to 8 in. (4) For low-grade coals from Montana, Utah and Washington, about 4 in.

The quantity of coal to be used in each firing (Henry Kreisinger "Firing Soft Coal Under Power-plant Boilers," *Government Printing Office*) depends upon the size of the grate and the intensity of the draft. When the total available draft in the uptake is about 1 in. of water, 2 to 2.5 lb. of coal fired per square foot of grate area is a fair average. Thus on a grate 8 ft. wide and 6 ft. deep, each firing would average 100 to 125 lb. of coal, or about 6 to 9 shovelfuls.

The intervals between the firings should be, on the average, about 5 min. long. If the draft is quite high, the periods may be shortened to 3 min. With a weak draft and sluggish fires, the interval may sometimes be lengthened to 8 min. Under ordinary circumstances it should never be longer than 10 min. Small and frequent firings make the coal supply more nearly proportional than otherwise to the air supply. The latter, with most hand-fired furnaces, is nearly constant. Small and frequent firings also tend to prevent the formation of a crust on the fire bed and the formation of holes therein.

403. Two general methods of hand-firing are commonly practiced: (1) the alternate or spreading method, (2) the coking method.

NOTE.—Hand-firing by the alternate method consists in spreading a charge of green coal over one certain portion of the grate area at one firing, and over another certain portion at the next firing. The idea is to continually preserve an area of incandescent fuel over which the volatiles distilled from the freshly fired fuel may pass. A three-door furnace (Fig. 293) may be alternately fired (Bureau of Mines Bulletin) thus: The portions A_1 and C_1 of the rear half of the grate area, which are respectively opposite doors A and C are, at one firing, charged through these doors. The portion of the front half $B1$ which is opposite door B is charged through door B. At the next firing, the end portions, A_2 and C_2, of the front half of the grate are, respectively, replenished through doors A and C, while the middle portion B_2 of the rear half is replenished through door B. The firing may often be done along alternate narrow strips extending from the furnace front to the bridge wall.

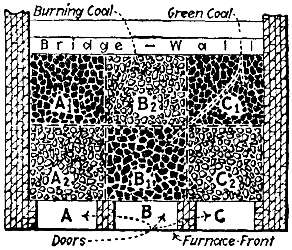


FIG. 293.—A method of alternate firing.

Hand-firing by the coking method consists in banking the charge of green fuel to a considerable depth on the front end of the grate. There, it remains until a large portion of the gases is burned out. The mass of partially coked fuel thus formed is then pushed back and spread over the grate. A bed of incandescent fuel is thus constantly maintained on the rear half of the grate. The gases from each succeeding charge of green fuel are ignited and burned while passing over this bed. It may be necessary to hold the fire doors slightly ajar in order to admit sufficient air for the combustion of the profuse volume of gases streaming from the banked up mass of fuel.

404. The relative advantages and disadvantages of the alternate and of the coking firing methods may be enumerated as follows:

With the alternate method: (1) higher efficiency of combustion may result; (2) the percentage of CO_2 may be greater; (3) the temperature of the flue gases may be lower; (4) steam generation may be more uniform; (5) more clinker and ash may accumulate in the furnace.

With the coking method: (1) the time intervals between successive firings are longer; (2) smoke may be abated; (3) fluctuating loads are handled with greater difficulty; (4) unseen holes may develop in the fuel bed; (5) the percentage

of CO_2 may be smaller. This may be due to the greater requirement for excess air.

405. Two general methods of cleaning a hand-fired furnace are available: (1) the side method (Figs. 294 and 295), (2) the front-to-rear method (Fig. 296).

NOTE.—In the side method of furnace cleaning (Figs. 294 and 295) one-half of the grate area, from the furnace front to the bridge wall, is cleaned at a time. The hoe may be used (Fig. 294) to scrape the coked fuel from the side which is to be cleaned first over to the opposite side.

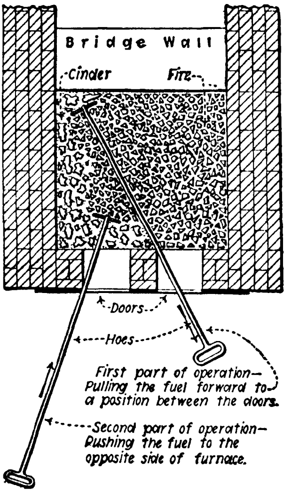


FIG. 294.—Side method of cleaning with hoe.

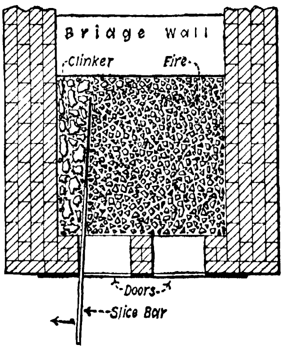


FIG. 295.—Side method of cleaning with slice bar.

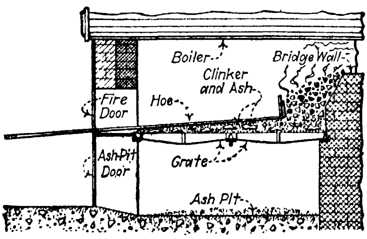


FIG. 296.—Front-to-rear method of cleaning fire.

A more-convenient and practical method is to use the slice bar (Fig. 295) for "winging" the fuel from the side which is to be cleaned over to the opposite side. The bared mass of furnace refuse is next broken up with the slice bar. It is then hoed out onto the floor. The burning fuel

is now winged over onto the bare half of the grate. Then enough green fuel to form a substantial bed is shoveled in on top of it. When the fire begins to burn intensely on the clean half of the grate, the remaining half is cleaned likewise.

In the front-to-rear method of furnace cleaning (Fig. 296) the coked fuel on one-half of the grate is pushed, with the hoe, back to the bridge wall. The mass of refuse thus exposed is then hoed out. The burning fuel is next pulled forward and leveled on the bare grate. The clean half of the fire is now replenished. When the fresh fuel is burning properly, the remaining half of the grate is similarly cleaned.

Thorough cleaning of a furnace is practically impossible with the front-to-rear method. This method is a makeshift which should be resorted to only under stress of an emergency, as when abnormal or unusual conditions of service might not allow the time necessary for proper cleaning by the side method.

QUESTIONS ON DIVISION 16

1. What is meant by combustible? Combustion?

2. Why is air necessary for combustion?

3. How is it possible to determine the completeness of combustion?

4. How much air is actually required for combustion of coal? How much is actually used in practice?

5. What general requirements must be met for proper combustion?

6. What makes proper combustion difficult?

7. How is anthracite fired?

8. Discuss a method of starting a soft-coal fire.

9. Give the general directions for firing soft coal.

10. Does the fuel-bed thickness vary for different coals? How much?

11. Discuss the two general methods of firing soft coal.

12. Compare, with reference to results obtained, the two general methods of firing.

13. How may thin spots in the fuel bed be eliminated?

14. What is the result of long-firing periods?

15. What happens if the fireman persists in spreading the coal evenly over the fuel bed?

16. How may the fuel bed be quickly leveled? Is this method advisable?

17. How often should soft coal be fired? Discuss.

18. Describe the proper standing position for the fireman.

19. Describe two methods of cleaning fires.

20. Compare the effectiveness of the two general methods of cleaning a fire.

DIVISION 17

STOKERS AND PULVERIZED COAL

406. The principal functions of stokers and pulverized-coal equipment (Figs. 297 to 320, inclusive) are: (1) to promote economical combustion by supplying the fuel uniformly to boiler furnaces; (2) to conserve labor; (3) to provide a means for effective combustion of low grades of coal which might otherwise be unavailable as fuel; (4) to facilitate smokeless combustion; (5) to increase combustion rates.

NOTE.—By feeding the fuel continuously to a boiler furnace, instead of intermittently, as with hand-firing, the supply of air may at all times be accurately regulated in conformity with the desired rate of combustion. This conduces to economy of fuel. In plants, with fuel consumptions of 200 tons or more per week, the labor costs with stokers might be from 30 to 49 per cent less than the labor costs with hand-firing. In plants showing smaller weekly fuel consumptions than 200 tons, the saving in labor cost with mechanical stoking might show a corresponding diminishment. In a plant using but 10 or 12 tons of coal per week, the saving in this item might be nil. Low grades of coal cannot, usually, be profitably burned in hand-fired furnaces. But properly selected stokers can successfully burn many poor grades of coal that cannot be burned on hand grates.

407. An ideal installation of fuel-burning equipment would be one capable of efficiently burning all types of fuel. No one system has yet been devised to meet this ideal, though combination oil, gas, and pulverized-coal burners come the closest to it. Most plants are so located with respect to transportation and proximity to fuel supply that their selection of fuels is limited economically to only a few grades. In such cases the ideal is not necessary, and that coal-burning equipment should be installed which will best handle the fuels economically available. The burning characteristics of the coal supply largely influence the type of equipment selected.

408. The different types of stokers may be classified under two general types—overfeed and underfeed. Overfeed stokers include hand stokers, sprinkler stokers, and traveling-grate stokers. Underfeed stokers include single-retort and multiple-retort types with various arrangements of ash removal. Overfeed stokers feed coal above the point of air supply; underfeed stokers supply coal under the point of air supply.

409. Hand-operated mechanical stokers (Fig. 297) possess some of the advantages of automatic mechanical stokers.

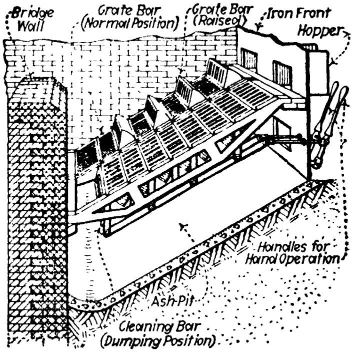


FIG. 297.—A hand stoker.

With a hand-operated stoker, the rated capacity of a boiler, as developed by hand-firing, may be exceeded by 30 to 75 per cent. The combined boiler-and-furnace efficiency may be improved. Cleaning of the fires may be rendered less difficult. Smokeless combustion may be secured. The cost of labor may be reduced.

NOTE.—Hand-operated stokers are especially adapted for small plants wherein mechanical stoking may be desirable, but where monetary considerations prohibit the installation of automatic stokers.

410. Sprinkler stokers are designed for firing small size coal. The fuel is sprinkled or showered on the grate. Distillation of its volatile constituents is presumed to be effected rapidly enough to prevent covering the grate with green coal, and thus the surface of the fire bed is continuously hot. Stokers of this type are usually confined to installations not over

10,000 sq. ft. of heating surface. Typical sprinkler stokers are shown in Figs. 298 and 299. Up to combustion rates of

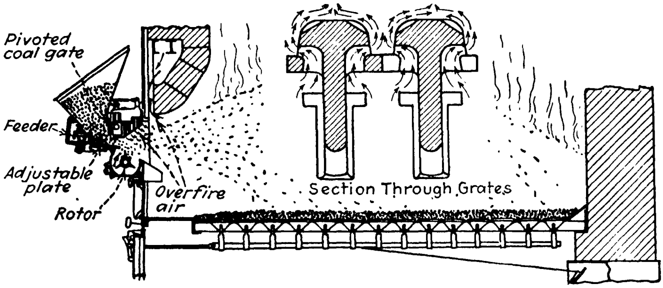


FIG. 298.—Sprinkler type of overfeed stoker with hand-operated shaking grates. (*Wm. Bros. Boiler & Manufacturing Company.*)

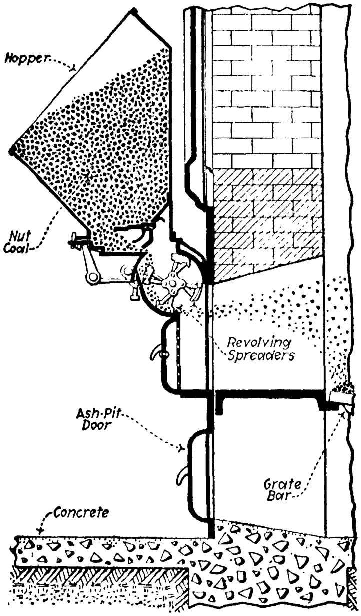


FIG. 299.—Sprinkling stoker with spreader turning under.

about 40 lb. per sq. ft. of grate area, these stokers handle successfully most kinds of coal.

411. Overfeed stokers of the inclined-grate type (Fig. 300) burn most grades of bituminous coal, though trouble may be experienced with coals having low ash content from grate burnouts or with ash having low fusion temperature forming clinker. These stokers usually operate with natural draft at not over 250 per cent rating. They are used in smaller installations with boilers having not over 6,000 sq. ft. of heating surface. The Wetzel stoker (Fig. 300) pushes the

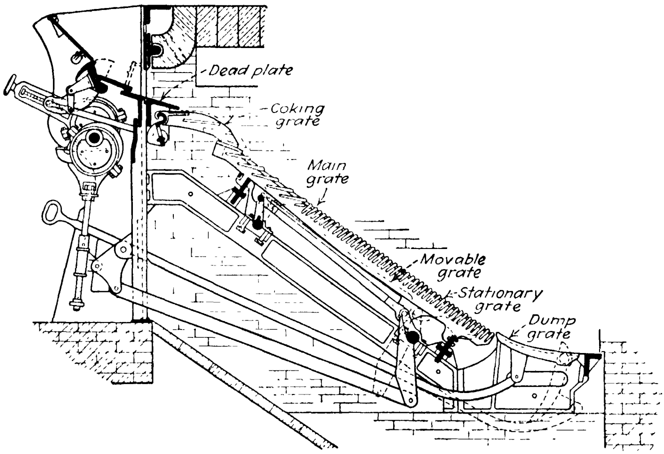


FIG. 300.—Inclined-grate overfeed stoker. (*Wetzel Mechanical Stoker Company.*)

coal from a hopper into the furnace and onto a nearly horizontal dead plate and then onto a coking grate that is given a backward and forward movement by the stoker mechanism. On this portion of the stoker, the volatile matter is distilled off and the coal ignited. A more or less extensive arch is sprung over this portion of the stoker and aids in the distillation of volatile and ignition. Secondary air to burn the volatile is supplied over the fire at A. The ignited coal moves by gravity and the action of the grates onto the main grate which has a reciprocating motion at right angles to the grate surface. Ash and clinker deposit on the dumping grate and are periodically dumped to an ashpit below.

412. Traveling-grate Stokers (Figs. 301 and 302) May Be of Either Chain-grate or Bar-grate Type.—In both types coal is fed from a hopper onto a continuously moving grate which carries the coal into and through the furnace. Depth of coal on the grate is controlled by an adjustable gate at the front of the stoker to a depth of 2 to 7 in. depending on the nature of the coal. The grate forms an endless chain driven

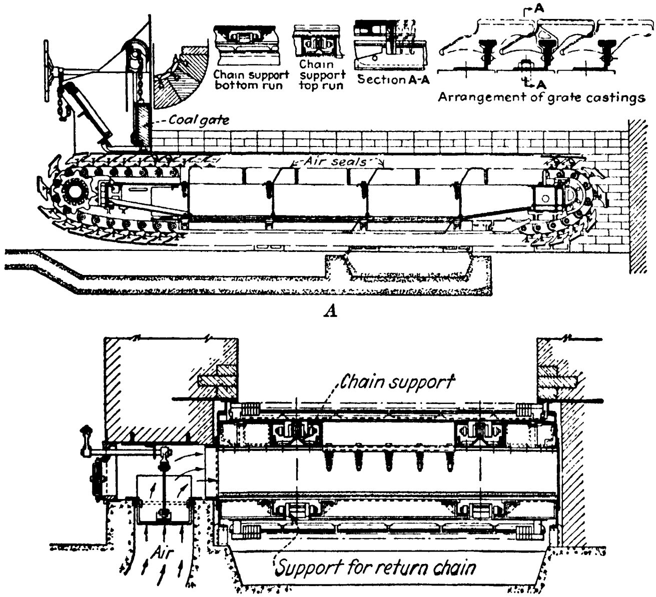


FIG. 301.—Bar-grate type of traveling-grate stoker. (*Riley Stoker Corporation.*)

by a sprocket at one end and passing over an idling drum or sprocket at the other. The bearings of the idler drum may be moved so as to adjust the tension in the grate. The driving sprocket is usually driven through worm and worm-wheel gears by either motor or steam engine. Speed of travel of the grate is varied according to the rate of combustion up to a maximum of about 90 to 100 ft. per hr., but 25 to 30 ft. per hr. is the average. Coal has been burned at the rate of

70 lb. per sq. ft. of grate an hour but usually the maximum rating is about 50 lb. per sq. ft. per hr. In chain-grate types the grate bars form the chain. In the bar-grate types the grates are carried on cross bars that are connected at each end to a steel chain. Chain grates are usually driven from the front end, and bar-grate stokers are driven from either the front or the ashpit end. Both types have been built up to 25 ft. wide and about 23 ft. long.

413. Air for combustion may be supplied by either natural or forced draft, but relatively few natural-draft stokers are

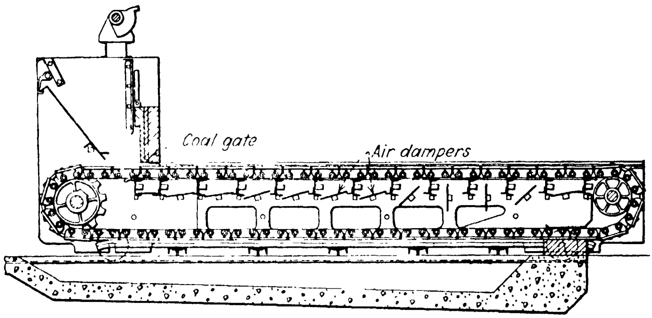


Fig. 302.—Natural-draft chain-grate stoker with front-drive sprocket. (*Illinois Stoker Company.*)

now being installed and these only in small sizes. Forced draft may be supplied from the side and between the two grate runs or from under the stoker so that the air passes through both grates. In either case the air supply is arranged in several individually dampered zones depending on the length of the stoker. It is important that air flow be controlled according to the rate of combustion at different parts of the grate and so that the coal is completely burned when it reaches the rear of the stoker and therefore ready to be discharged as refuse.

414. Traveling-grate Stokers Are Designed to Burn Free-burning Noncoking Coals.—The grate-bar type is particularly designed for burning the small sizes of anthracite and coke breeze and will burn successfully grades of free-burning bituminous coals that have an ash which does not stick to the

grates. The chain-grate stoker is particularly adapted for burning noncoking bituminous coals, as the grate bars have a self-cleaning action in passing over the rear drum that breaks loose any clinker or ash that tends to stick. They will also burn anthracite and coke breeze. Both types will burn the western subbituminous and lignites.

415. Underfeed Stokers (Figs. 303 to 308) Feed the Fire Bed from the Under Side.—The coal is fed from an overhead bunker by gravity to a stoker hopper. Rams or plungers

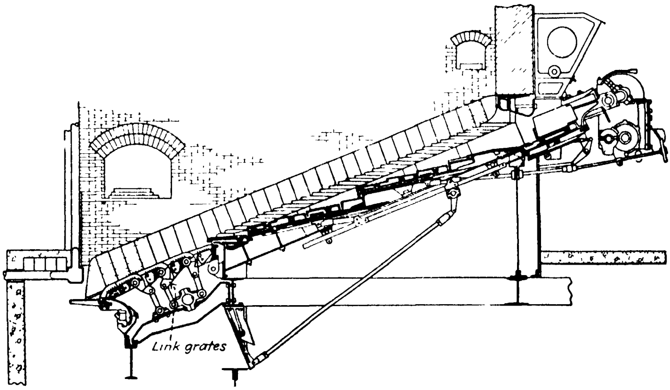


FIG. 303.—Underfeed stoker with continuous ash discharge and link extension grates. (*Westinghouse Electric and Manufacturing Company.*)

driven by the stoker mechanism push the coal into horizontal or inclined retorts. On both sides of the retorts are tuyères through which forced draft supplies air for combustion. Underfed stokers may have one or more retorts as required by the size of the boiler. Secondary rams or pushers keep the fuel bed broken up, feed the coal to the ash-removal end of the stoker, and aid in removal of clinker. In multiple-retort stokers the ash refuse is fed onto dump grates which may be lowered periodically dumping the ash into a pit. In larger central-station installations the ash is fed to a pit from which revolving clinker grinders remove the refuse to an ash hopper. In the stoker in Fig. 303, an articulated link-extension grate feeds the refuse continuously through an opening formed by an adjustable apron and a water back.

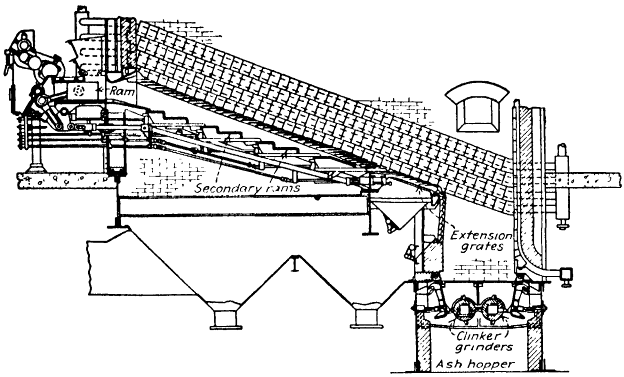


FIG. 304.—Underfeed stoker with clinker grinder rolls for ash removal. (*Combustion Engineering Company.*)

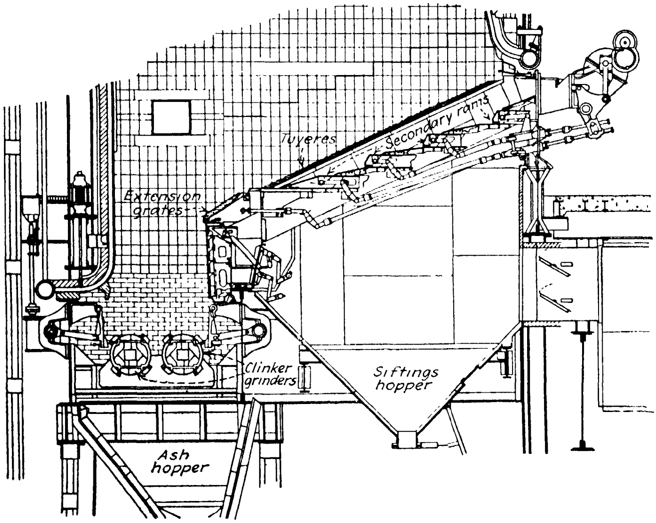


FIG. 305.—Taylor multiple-retort underfeed stoker with clinker grinders four secondary pushers and reciprocating extension grates. (*American Engineering Company.*)

416. The rams of the majority of underfeed stokers are operated by cranks driven through reduction gears by motor, engine, or turbine. Shear pins or other safety devices are provided to protect the stoker drive from breakage in case foreign matter jams in the stoker. Two or more speeds

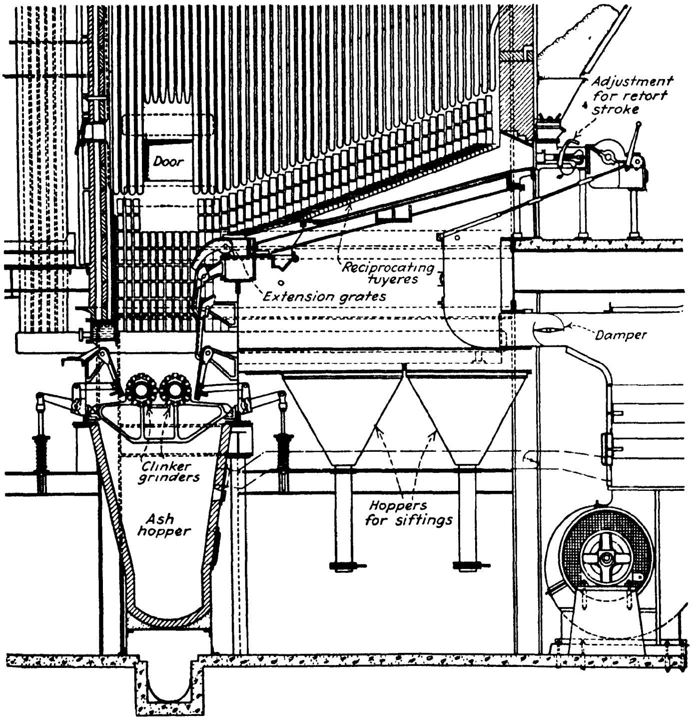


FIG. 306.—Multiple-retort underfeed stoker with reciprocating retort sides and tuyères. (*Riely Stoker Corporation.*)

are provided in the reduction gear. Multiple-retort stoker drives are divided into sections, each with its own reduction gears driven from a line shaft. This makes it possible to increase or decrease the coal feed to individual sections as required by the condition of the fire. The same crank mechanism operates the secondary pushers which are usually

arranged so the length of their stroke can be varied. The stoker of Fig. 306 does not have secondary rams but the tuyère stacks are divided in half lengthwise, and each half reciprocates in opposite directions feeding the coal to the rear of the stoker and keeping the fuel bed broken up. Some stokers are arranged with rams driven hydraulically or by steam cylinder.

417. Air for combustion is delivered by the forced-draft fan to a wind box under the stoker. Dampers are arranged

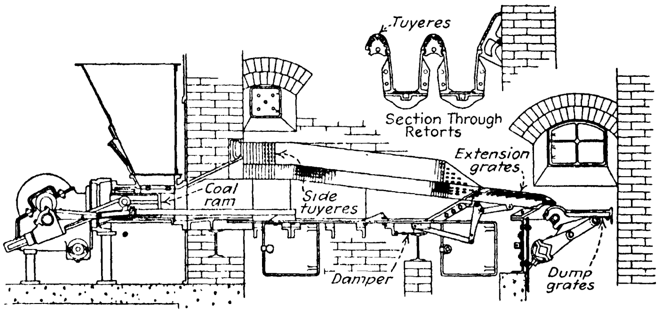


Fig. 307.—Multiple-retort underfeed stoker with power operated dump ash grates. (*Detroit Stoker Company.*)

under the tuyère stacks so that the air to each stoker section can be regulated and so prevent excess air from blowing through holes in the fuel bed. In a few large installations, the stoker has been divided into a large number of zones with metering dampers that automatically control the air to each stoker section or that indicate excessive air flow on a control panel and permit manual adjustment by the fireman. Such arrangements are expensive and complicated, but have increased combustion rates per square foot of projected grate area to 78 lb. per hr., an increase of 23 per cent over previous maximum. (For a complete description see the paper by A. S. Griswold and H. E. Macomber, "Distribution of Air to Underfeed Stokers" presented to A.S.M.E., December, 1935.) Without zoned control continuous coal-burning rates are limited to about 55 lb. per sq. ft. of projected grate area per hour.

418. Underfeed stokers are designed to cause some agitation of the fuel bed and hence are particularly successful in burning bituminous coals that have coking or matting characteristics. Coke masses prevent flow of air through the bed unless broken up. Coals that have ash with low fusion temperature are apt to give clinker trouble and make ash removal difficult. A recently developed stoker with water-cooling tubes recessed in the tuyères has been reported as successfully burning Iowa coals with 10 per cent ash having a fusing temperature of 1900° at a combustion rate of 54 lb.

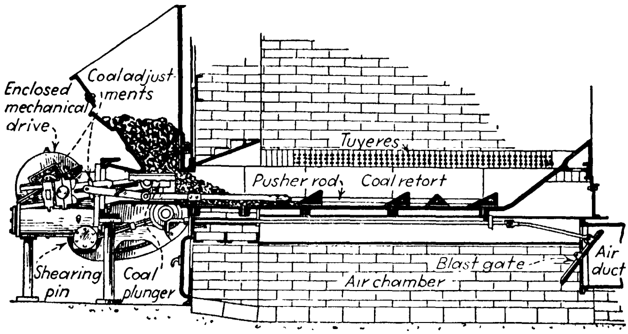


FIG. 308.—Double-retort underfeed stoker, side ash removal. (*Detroit Stoker Company.*)

per sq. ft. of air-admitting surface per hour. Underfeed stokers are applicable to very large boilers and have evaporated over 500,000 lb. of steam an hour. The largest stoker of this type built has 15 retorts 69 tuyères long and 694 sq. ft. of projected grate area. Efficiency on test of this stoker were reported as 89 per cent at low load and 77.5 per cent at maximum load of 504,000 lb. per hr. with a boiler having 24,450 sq. ft. of heating surface equipped with a 22,400 sq. ft. economizer.

419. Single-retort stokers (Figs. 308 and 309) for use with boilers up to about 5,500 lb. of steam per hour capacity have been developed by many manufacturers. They are nearly all arranged to feed coal from a hopper by a screw conveyer into a retort. Around the top of the retort are tuyères that supply air for combustion. Coal fills up the retort and spreads over the tuyères. The ash falls off to the sides.

420. Burning pulverized coal has become increasingly important both in central stations and industrial plants.

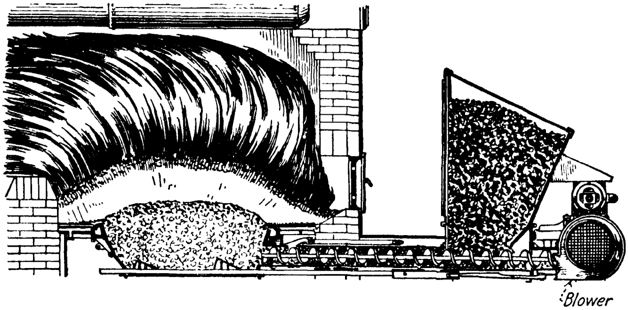


FIG. 309.—Underfeed stoker with fuel fed by screw. (*Iron Fireman.*)

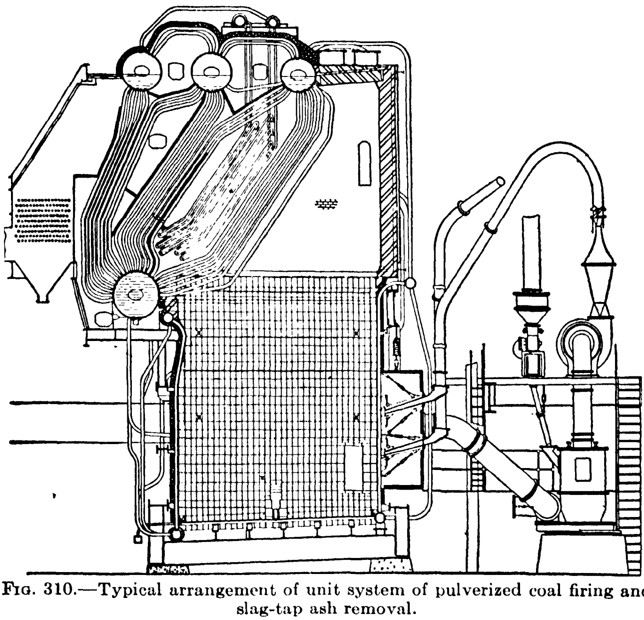


FIG. 310.—Typical arrangement of unit system of pulverized coal firing and slag-tap ash removal.

Practically all grades of bituminous coal and some anthracite can be burned in this form both economically and efficiently. A somewhat closer control of excess air is possible with this

method of firing than with many stokers, though in the larger stoker installations equal efficiency is obtainable. It offers greater freedom in coal purchases, and furnaces and burners designed for pulverized coal can be readily changed over for oil or gas. Certain coals are particularly abrasive and hard to pulverize and hence cause excessive maintenance on pulverizing equipment.

421. There are two systems of pulverized coal firing—bin system (Fig. 311) and unit or direct firing (Fig. 310). In the bin system the coal is pulverized and then delivered to boiler bunkers. From the bunkers it is fed as required into a stream of primary air that carries it through piping to the burners. In the unit system the coal is fed to the pulverizing mill as required and delivered by primary air direct to the coal burners. In both, combustion of the coal takes place while it is in suspension in the furnaces. The bin system has been used to considerable extent in large central-station plants where it offers some advantages. Industrial plants use the unit system, it being less complicated. With increasing mill capacity central stations have also adopted the unit system in many cases.

422. The fineness to which these mills pulverize the coal is determined by passing a sample of the coal through screens

TABLE XIII.—ECONOMIC FINENESS TO WHICH VARIOUS COALS SHOULD BE PULVERIZED

Coal	Percentage through		
	200 mesh	100 mesh	50 mesh
Eastern low volatile	75	92	98
Eastern high volatile	65	85	97
Illinois	55	80	92
Subbituminous	45	75	90
Lignite	35	65	88
Anthracite	85	97	100

of various sizes and measuring the weight left on each screen from which the percentage of the coal passing through can be calculated. Thus 80 per cent through a 200-mesh screen

means that this percentage of the weight of the coal will pass a standard screen having 200 openings to the inch. For best

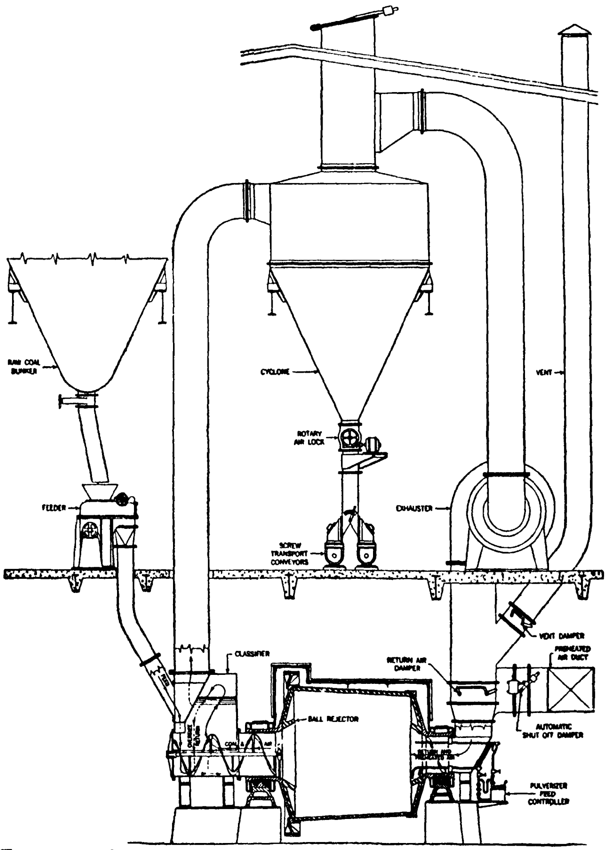


FIG. 311.—Typical arrangement of pulverized coal-preparation equipment for a bin and feeder, or storage system. The pulverized coal is delivered to bunkers near the boilers. The ball mill (at the bottom of the figure) is partly filled with steel balls (not shown) which grind and pulverize the coal as the mill is rotated. (*Foster Wheeler Corporation.*)

economic results the fineness to which various coals should be ground is given in Table XIII taken from a paper by

Kreisinger presented before the Chicago Fuels Meeting, *A.S.M.E.*, Feb. 10–13, 1931.

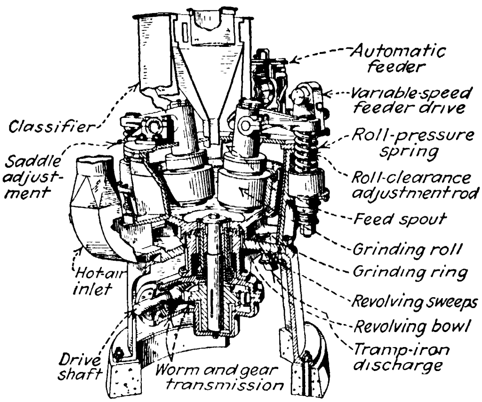


FIG. 312.—Bowl mill pulverizer with air classifier. (*Combustion Engineering Company.*)

NOTE.—In this pulverizer coal is ground between rollers and the sides of a revolving bowl. Owing to the slanting sides of the bowl and centrifugal force the ground coal works up the sides of the bowl. Fines and intermediate sizes are picked up at the top edge of the bowl by a current of air and carried into a classifier above for separation. The fines are carried out of the mill by the air current while the coarser coal drops back into the bowl for further pulverizing. Pressure of the rolls against the sides of the bowl is controlled by springs which are adjustable from outside the mill, as is also the clearance between the rolls and the sides of the mill.

423. Pulverizing is done by (1) impact, (2) grinding, (3) attrition or a combination of these. Figures 312 to 315 illustrate various types of pulverizers used in power-plant practice. All of these employ a current of air (often preheated so as to dry the coal) to separate and carry out of the mill the coal that has been reduced to the proper size. In the case of unit systems this air stream carries the pulverized coal to the burner and blows it into the furnace. In bin systems the air and coal mixture from the mill passes through a cyclone which separates the coal from the air and delivers it to the transport system that discharges to the boiler bunkers.

Pulverizers vary in size up to a maximum of about 50 tons per hr.

424. Pulverized-coal burners have many forms, some designed for vertical firing, others for horizontal firing, some designed to fire between the tubes of a waterwall, others circular in shape for installation in a refractory wall. Figures 316 to 319 illustrate these various types as well as combination coal, oil, and gas burners. In all of them provision is made for supplying secondary air, either under forced or natural draft, in such a way as to mix thoroughly with the stream of primary air and coal.

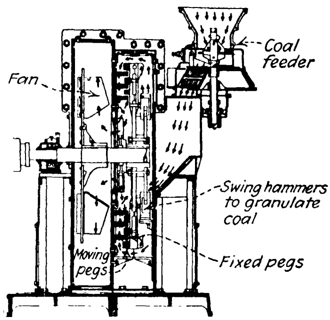


FIG. 313.—Combination impact and attrition pulverized coal mill. (*Riley Stoker Corporation.*)

NOTE.—Coal is fed by the feeder to the first stage of pulverization, which consists of a series of swing hammers. The raw coal is broken down by impact with these hammers to a granular state and then passes to the final or attrition stage around the outside of the rotor. Here moving pegs fastened to a revolving disk travel between stationary pegs attached to the mill housing. In this stage the coal which is traveling at high velocity is pulverized by friction against itself and the mill surfaces. From the attritor stage the coal is drawn to the center of the mill through a series of rapidly rotating rejector arms and into the fan chamber. The rejector arms throw the coarse particles back into the maze of revolving and stationary pegs for further pulverizing. The air current produced by the fan picks up the fine coal and delivers it direct to the coal burner. At maximum load about 10 per cent of the air required for combustion is supplied by the pulverizer.

425. Ash from pulverized coal collects in the furnace bottom from which it may be removed by hydraulic ash-removal systems or hoed out by hand. In some installations the ash is maintained in molten state. The fluid ash is tapped off periodically and quenched by streams of water which impinge against it as it issues from the furnace. The water breaks the ash up into gravellike particles. This method of ash removal is particularly applicable when the ash has a low

fusion temperature. With *slag-tap ash removal,* as it is called, the furnace bottom is usually water-cooled or provided with specially shaped refractory arranged to permit expansion without pushing against the walls of the furnace. Plastic chrome is used to considerable extent in the bottoms of slag-tap furnaces.

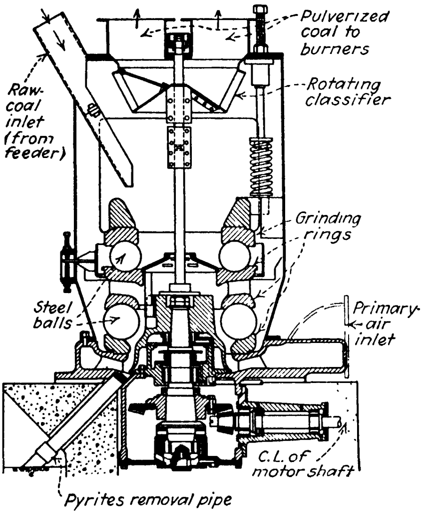


FIG. 314.—Ball and grinding-ring type of pulverizer. (*Babcock and Wilcox Company.*)

NOTE.—Coal fed into the top of the mill is ground and pulverized between steel balls and ball races or grinding rings. The center grinding ring is rotated by the vertical shaft and causes the two rows of balls to rotate with it. Pressure between the balls and the grinding rings is regulated by springs. Air entering the bottom of the mill carries the fines up to a classifier and then out of the mill to a cyclone separator if used with the bin system or direct to coal burners if a unit system. Smaller mills of this type have one instead of two ball races.

426. A considerable portion of the pulverized-coal ash is carried out of the furnace with the flue gas, and unless caught by dust collectors it is discharged from the stack and may become a nuisance to the surrounding neighborhood. Various types of dust collectors are available and include dry types

that depend on change of direction or centrifugal force for their operation, wet types, and electrostatic precipitators.

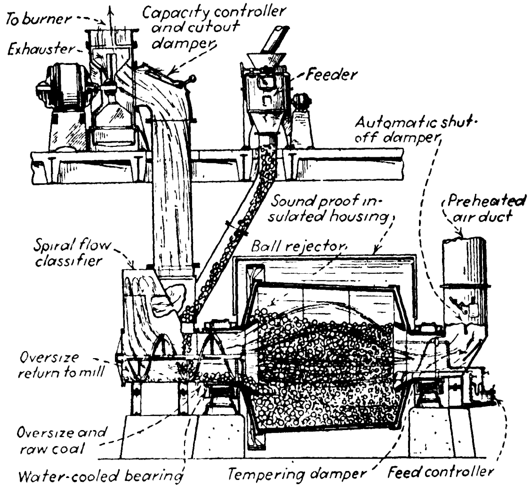


FIG. 315.—Foster Wheeler Tricone ball mill (Hardinge type) arranged for unit system of firing with exhauster and feeder on floor above mill. The mill and classifier are shown in section to illustrate method of operation and the approximate distribution of the material and ball charge within the mill. In this illustration the mill is shown turning with the top moving toward the reader. Thus the charge of fuel and balls is carried up the far side of the mill whence it cascades toward the bottom, with pulverizing effect. Air sweeping through the mill carries fines to the to the burner.

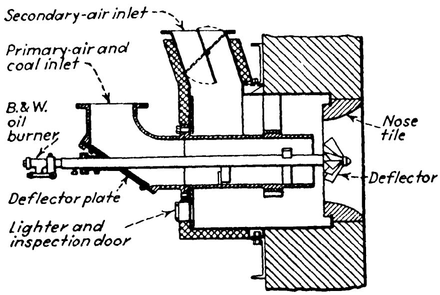


FIG. 316.—Circular burner for pulverized coal. (*Babcock and Wilcox Company.*)

427. Danger from fire or explosion exists with pulverized coal as with any combustible dust. The National Board of

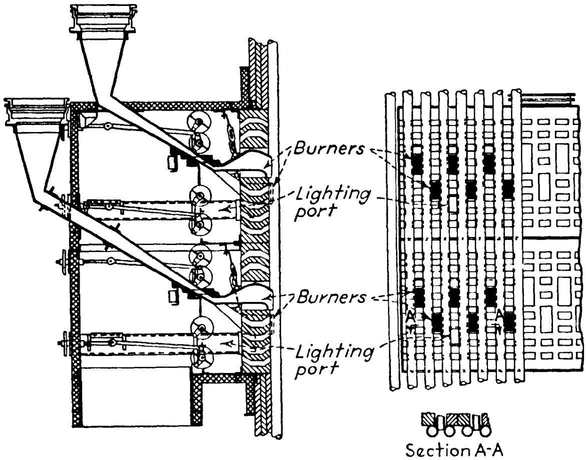


Fig. 317.—Cross-jet or intertube type burner for firing pulverized coal horizontally through a water-cooled wall. The burner, as shown, is designed for force-draft operation but similar burners are furnished for natural-draft operation and for firing through refractory walls. (*Foster Wheeler Corporation.*)

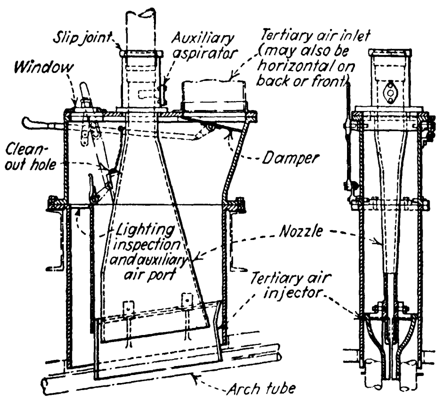


Fig. 318.—Pulverized coal burner for vertical firing through arch. (*Combustion Engineering Company.*)

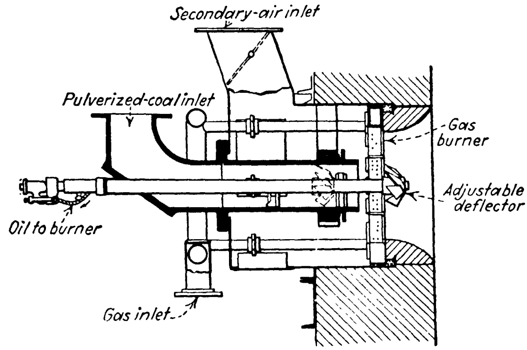


FIG. 319.—Circular burner for pulverized coal, oil, and gas. (*Babcock and Wilcox Company.*)

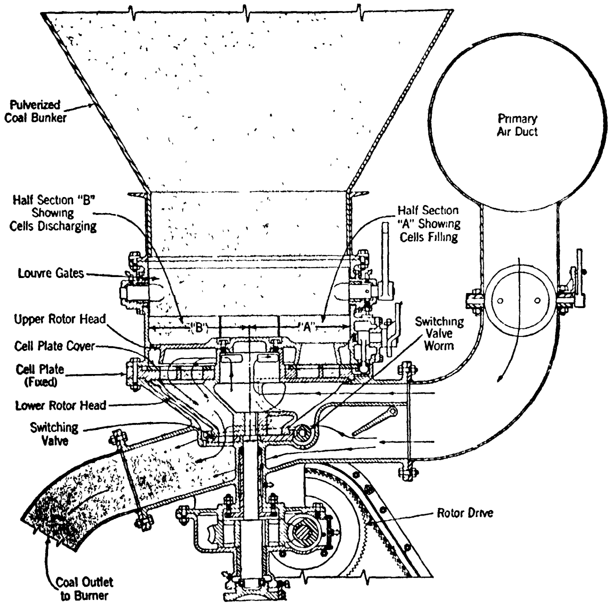


FIG. 320.—Pulverized-coal feeder used with the bin-storage system of firing. (*Foster Wheeler Corporation.*)

Fire Underwriters of New York City have made regulations for the installation of pulverized-coal system, which are approved by the American Standards Association and if followed will reduce the hazards to a minimum. The main thing is good housekeeping, *i.e.*, keeping all pulverized-coal handling equipment dusttight, not allowing coal dust to accumulate, and wetting it before sweeping. Eliminate all sources of ignition near pulverizing or transportation equipment and provide means for preventing tramp iron from getting into the pulverizer.

QUESTIONS ON DIVISION 17

1. What functions must mechanical-firing equipment perform?
2. What are the requirements of an ideal fuel-burning system?
3. Name the different types of stokers.
4. Explain the difference between overfeed and underfeed firing.
5. Name two types of overfeed stokers. Describe them.
6. Name two types of traveling-grate stokers. What is the difference between them?
7. What kinds of coal may be fired on traveling grates?
8. What trouble is likely to result if coal with low-ash content is fired on traveling grates?
9. What depth of fuel bed is usually carried?
10. What fuel-burning rates are possible on chain grates? On bar grates?
11. Which type of traveling grate is designed for burning anthracite? Which for coke breeze?
12. How is coal fed in underfeed stokers?
13. What means are provided for distributing the coal over the length of the stoker?
14. What are tuyères?
15. How are the rams of underfeed stokers driven?
16. Can the stroke of the main ram be changed? Is it possible to adjust the movement of the secondary rams?
17. How is air supplied to the underfeed section?
18. What burning rates are possible with underfeed stokers?
19. How are ashes handled?
20. Describe two methods of continuous ash disposal.
21. What kind of coal should be burned on underfeed stokers?
22. Why will coal with ash of low fusion temperature give trouble?
23. What are the clinker grinders? How do they operate?
24. How is air to the different sections of the stoker controlled?
25. What is zoned air control? In what ways does it improve stoker operation?

26. What has made it possible to burn coal with low-fusion ash on underfeed stoker?

27. What method of coal feed is employed in single-retort stokers?

28. What are two systems of pulverized-coal firing? Describe them.

29. How fine is coal ground to burn in pulverized form?

30. How is coal pulverized?

31. Describe two types of pulverizers.

32. How is coal fed to the pulverized coal burners?

33. Where is air for combustion introduced with pulverized-coal firing?

34. How is ash removed from a pulverized-coal-fired furnace?

35. What is a slag-tap furnace?

36. Why are dust collectors used with pulverized coal?

37. Does any hazard exist with pulverized coal?

DIVISION 18

PETROLEUM AND GASEOUS FUELS

428. Petroleum is a fossil oil, presumably of either animal or vegetable origin. It is found in pockets in the earth's interior. It is the only oil which, by reason of its abundance and comparatively low cost, can be profitably used for fuel in boiler plants.

In recent years oil has become increasingly popular as a boiler fuel in both industrial and central-station power plants.

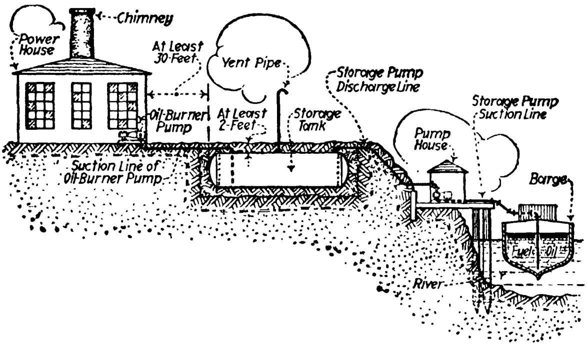


FIG. 321.—Conveyance and storage of fuel oil

Boiler fuels are generally classified according to navy specifications as Bunker *A*, Bunker *B*, and Bunker *C* oils. Of these Bunker *C* oil is the cheapest, and is the fuel usually used. It has a viscosity of not over 300 sec. Saybolt Furol at 122°, a density of 5 to 14° A.P.I., and a heating value of about 18,500 B.t.u. per lb. (See Table XIV for heating value and density of oils.) Other important properties of fuel oil are water content which should not be over 2 per cent and sediment which should be not greater than 0.25 per cent.

NOTE.—The petroleums from the eastern oil fields of the United States contain a paraffin base. Their distillates are such that they are more

valuable for other purposes than as boiler fuel. The petroleums from the western and some of the southern oil fields contain an asphaltum base. These, either refined or in their crude or natural state, mainly supply the bulk of the steam-making fuel oil which is used in the United States.

429. The chief advantages of petroleum fuel are: (1) its low cost of conveyance and storage—oil can be delivered (Fig. 321) into storage tanks at a comparatively trifling cost; (2) relatively small storage room which it requires—approximately 50 per cent more potential heat energy can be stored with oil than with coal in a given space; (3) the

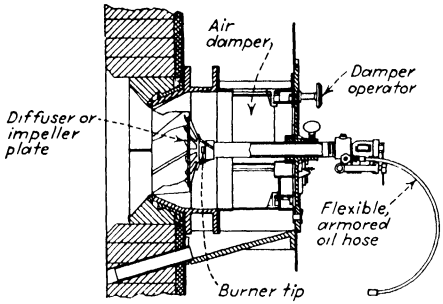


FIG. 322.—Complete assembly of mechanical oil-burner register and tip. (*Babcock and Wilcox Company.*)

more nearly perfect combustion which it affords—oil requires less excess air for its combustion than does coal, leaves no ash, and facilitates maintenance of a constant furnace temperature and elimination of smoke; (4) its immunity from deterioration of calorific value while in storage; (5) its immunity from spontaneous combustion while in storage; (6) the ease with which its intensity of combustion can be regulated to meet load fluctuations; (7) the saving of labor which its use affords; (8) the comparative cleanliness which attends its use.

430. The chief disadvantages of petroleum fuel are: (1) restrictions, with respect to locations of oil tanks, which are imposed by insurance and municipal regulations; (2) liability of injury to boiler plates by local concentrations of its heating effect; (3) its tendency to distill explosive vapors. But to

explode the vapor it must be ignited. This should not be confused with spontaneous combustion.

TABLE XIV.—OIL MEASUREMENT AT 60°F.*

Gravity, °A.P.I.	Specific gravity	Lb. per gal.	B.t.u. per lb.	B.t.u. per gal.	Lb. per 42-gal. barrel	Weight, lb. per cu. ft.	Cu. ft. per ton 2,240 lbs.	Gals. per ton of 2,240 lb.
3	1.0520	8.76	18,190	159,340	368.00	65.54	34.17	255 65
4	1.0443	8.69	18,240	158,500	365.31	65.07	34.42	257.54
5	1.0366	8.63	18,290	157,840	362.62	64.59	34.68	259.48
6	1.0291	8.57	18,340	157,170	359.98	64.12	34.93	261.37
7	1.0217	8.50	18,390	156,320	357.37	63.65	35.18	263.26
8	1 0143	8.44	18,440	155,340	354.81	63.19	35.44	265.15
9	1.0071	8.39	18,490	155,130	352.46	62.78	35.68	266.91
10	1.0000	8.33	18,540	154,620	350.15	62.36	35.91	268.67
11	0.9930	8.27	18,590	153,740	347.71	61 93	36 16	270 56
12	0.9861	8.22	18,640	153,220	345.28	61 50	36 42	272.50
13	0.9792	8.16	18,690	152,510	342.88	61.07	36 67	274 39
14	0.9725	8.10	18,740	151,790	340.53	60.65	36.93	276.28
15	0.9659	8.05	18,790	151,260	338.22	60.24	37.18	278.17
16	0.9593	7.99	18,840	150,530	335.91	59.83	37.43	280.06
17	0.9529	7.94	18,890	149,980	333.64	59.42	37.69	281.99
18	0.9465	7.89	18,930	149,360	331.42	59.03	37.94	283.88
19	0 9402	7.83	18,980	148,610	329.23	58 64	38 19	285 77
20	0.9340	7.78	19,020	147,980	327.05	58.25	38.45	287.66
21	0 9279	7.73	19,060	147,330	324.91	57.87	38 70	289 55
22	0.9218	7.68	19,110	146,760	322.81	57.49	38 95	291.44
23	0.9159	7 63	19,150	146,110	320.71	57.12	39 21	293.37
24	0 9100	7.58	19,190	145,460	318.65	56 75	39 46	295.26
25	0.9042	7 53	19,230	144,800	316.59	56.39	39.72	297.15
26	0.8984	7.49	19,270	144,330	314.58	56.03	39.97	299.08
27	0.8927	7.44	19,310	143,670	312.60	55.68	40.22	300.97
28	0.8871	7.39	19,350	142,990	310.63	55 32	40 48	302.86
29	0.8816	7.35	19,380	142,440	308.70	54.98	40.73	304.75
30	0.8762	7 30	19,420	141,770	306.81	54.64	40.98	306.64
31	0.8708	7.26	19,450	141,210	304.92	54.31	41.24	308.53
32	0.8654	7.21	19,490	140,520	303.03	53.97	41.50	310.46
33	0.8602	7.17	19,520	139,960	301.18	53.64	41.75	312.35
34	0.8550	7.12	19,560	139,270	299.37	53.32	42.00	314.24
35	0.8498	7.08	19,590	138,690	297.57	53 00	42.26	316.18
36	0.8448	7.04	19,620	138,120	295.80	52.68	42.51	318 07
37	0.8398	7.00	19,650	137,550	294.04	52.37	42.76	319.96
38	0.8348	6.96	19,680	136,970	292.32	52.06	43.02	321 85
39	0.8299	6.92	19,720	136,460	290.64	51.76	43 26	323.69
40	0.8251	6.87	19,750	135,680	288.91	51.46	43.52	325.63
41	0.8203	6 83	19,780	135,090	287.23	51.16	43.78	327.52
42	0 8156	6.79	19,810	134,510	285.55	50.86	44.04	329.45

NOTE.—The above relation between specific gravity and A.P.I. degrees is expressed by the formula $\frac{141.5}{131.5° + \text{A.P.I.}°}$ = sp. gr. at 60°F.

For each 10°F. above 60°F. add 0.7°A.P.I.

For each 10°F. below 60°F. subtract 0.7°A.P.I.

* From *Power*.

431. Six Factors That Are Important to a Satisfactory Oil-firing Installation.—These are (1) furnace, (2) atomization, (3) air and fuel mixture, (4) range of rating, (5) draft, (6) distribution of fire. The question of furnace design is

covered in Div. 19. It should be said, however, that the furnace and kind of boiler influences the type of burner selected and its location as will be discussed later. Adequate draft must be available to draw the air required to burn the oil through the burner regulators or other openings. If this is not adequate, the output of the boiler will be limited unless forced draft is used. The remaining factors are largely influenced by burner design and should be considered and investigated when selecting the type of burner for any given installation.

432. Burners Are Classified According to the Method of Obtaining Atomization.—There are three types: (1) mechanical, (2) steam or air, (3) rotary cups. Typical examples of these types are illustrated.

Mechanical burners (Fig. 322) are supplied with oil at 200 to 300 lb. pressure when operating at maximum output. The oil must be heated so its viscosity is not over 130 to 200 sec. Saybolt Universal which with Bunker C oil requires temperature of 180 to 220°. Many burner tips are designed to impart a high-velocity rotary motion to the oil in a chamber before the orifice through which the oil issues into the furnace. The high pressure and rotary motion serve to break the oil up into a fine mist. Figures 323 to 325 show typical burner tips. In these tips the oil passes through tangential slots to a circular chamber, the combination imparting the rotary motion.

433. Air for combustion passes through registers provided in burners of this type. Guide vanes are usually provided that give uniform distribution of air around the burner and also impart a rotary motion to the air. These vanes are adjustable and are set to give a flame shape best suited to the local furnace condition. Diffuser plates of various shapes are placed just behind the tip and serve to control flame shape and flow of air. One of their main functions is to prevent direct impingement of the air flow from blowing out the flame before proper ignition has been obtained. The position of the diffuser plates is adjustable axially and is usually set so the flame fills the burner throat thus forcing air coming in through the register to pass through the flame

before entering the furnace. Mechanical burners usually produce a conical-shaped flame.

434. Output of the boiler with mechanical burners is controlled by changing the oil pressure at the burner tip, and when the capacity of the spray plates being used are exceeded

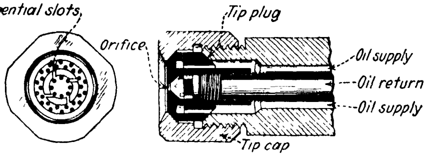


FIG. 323.—Wide-range mechanical-atomizing oil-burner tip with return oil control. (*Peabody Engineering Company.*)

by cutting out burners or changing burner tips. Unless special provisions are made the range of operation possible by changing the oil pressure is on the order of 1.5 to 1. The rate of oil flow through a burner varies as the square root of the ratio of the pressure so a drop from 200 to 50 lb. pressure reduces the rate of oil flow and hence the output by only one-half. At this reduced pressure atomization is appreci-

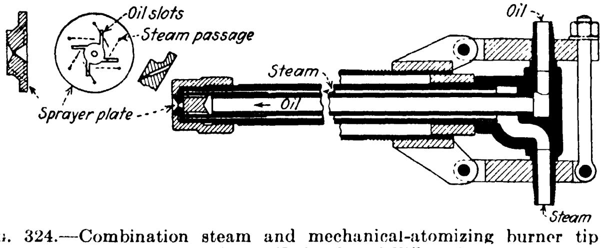


FIG. 324.—Combination steam and mechanical-atomizing burner tip for wide-range operation. (*Babcock and Wilcox Company.*)

ably coarser interfering with proper combustion. Several burners have been designed to increase the operating range to about 4 to 1 without seriously affecting atomization. Oil supply to the tip (Fig. 323) is maintained at 200 lb. for all loads. At full capacity no oil is allowed to flow from the return connection. By increasing the quantity of oil flowing to the return line, less oil flows through the orifice to the furnace thus reducing boiler output. But the maximum quantity of oil is flowing through the tangential slots at all

loads maintaining the high-velocity rotary motion behind the orifice and thus maintaining atomization over a wider range. Another burner (Fig. 324) is arranged so that steam atomization may be used for low-load operation. In Fig. 325 a primary and secondary oil supply is employed, the secondary supply being cut out at low loads.

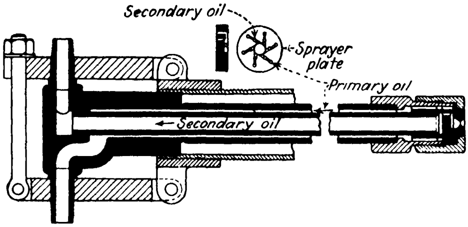


FIG. 325.—Wide-range mechanical-atomizing oil-burner tip with primary and secondary oil supply. (*Babcock and Wilcox Company.*)

435. Changing burner tips is something of a nuisance though used to some extent. Cutting out burners is a method frequently used in multi-burner installations and provides great flexibility. However, care should be taken to close the register, pull back the diffuser, and remove the tips from the burners that are not operating, as otherwise oil on the burner tip may carbonize making it necessary to clean it and air leakage will occur through an open register decreasing efficiency.

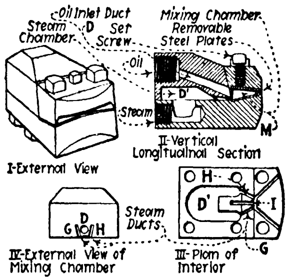


FIG. 326.—Details of tip of inside-mixing steam-atomizing oil burner.

436. Capacity of mechanical oil burners runs from 15 to 1,100 gal. of oil per hour. Forced draft is usually necessary with the higher capacity burners because the resistance of the burner register is usually too high for natural draft as produced by the average chimney. With forced draft it is possible to maintain more nearly balanced pressure in the furnace, and hence air leakage through the boiler setting is less.

437. There are two general types of steam-atomizing burners: (1) the inside-mixing type with which the oil and steam mingle within the burner; (2) the outside-mixing type with which the oil and steam meet after issuing from the burner. In both types the only purpose of the steam is to

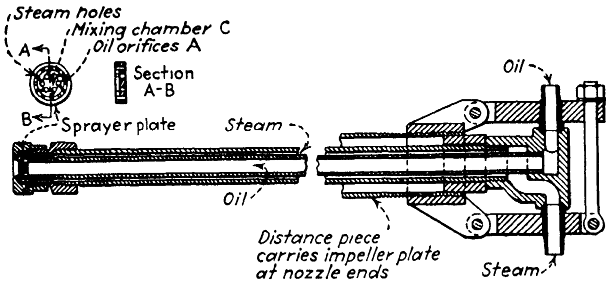


FIG. 327.—Inside-mix steam-atomizing conical flame oil-burner barrel assembly used in standard Babcock and Wilcox register.

atomize the oil which is accomplished by directing the steam so it cuts across the path of oil flow.

438. The inside-mixing burner shown in Fig. 326 gives a flat flame. Steam supplied at D cuts across the oil flow at M, mixes thoroughly with it, and atomizes the oil as it expands through the rectangular orifice. In the burner tip shown by Fig. 327 steam enters the mixing chamber C through tan-

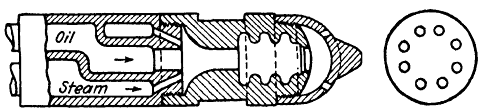


FIG. 328.—Inside-mix steam-atomizing oil-burner tip. (*Coen Company.*)

gential slots and cuts across the oil entering at A. Figure 328 is a similar type. The rotating motion set up thoroughly mixes the oil and steam and the mixture passes through orifice holes in the tip. These can be arranged to give either flat or conical flame. This burner tip when arranged with a conical flame is inserted in the same type of register used by the mechanical tip burner of Fig. 322.

439. Outside-mixing Burners (Figs. 329 and 330).—The principle of operation is much the same as the inside-mixing

type except that the oil and steam mix outside the burner tip. This type of burner is particularly applicable when heavy refinery sludges are to be burned.

440. Steam-atomizing burners require dry steam at not less than about 30 lb. per sq. in. pressure though some burners have operated at steam pressures as low as 10 lb. The

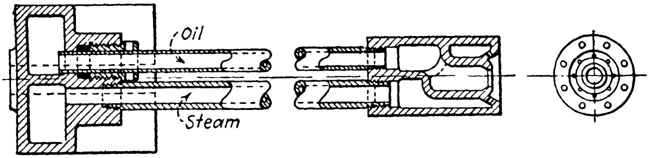


FIG. 329.—Outside-mix steam-atomizing oil burner for refinery sludge. (*Coen Company.*)

pressure under which fuel oil should be delivered to a steam-atomizing burner depends upon the viscosity of the oil and the type of burner. It may range from about 5 to over 70 lb. per sq. in. With close attention, steam consumption may be as low as 1 per cent of the total steam generated, but usual operation is between 2 and 3 per cent. Steam consumption

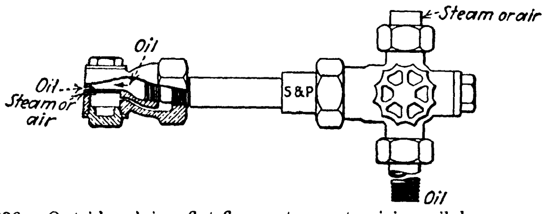


FIG. 330.—Outside-mixing flat-flame steam-atomizing oil burner. (*Staples and Pfeiffer, Ltd.*)

of more than about 5 per cent represents poor operation. If steam supply is deficient, a smoky or sparking flame may result but the same indications may be caused by too little air.

441. Steam burners generally are not made with as large capacity as mechanical burners, about 250 gal. per hr. being the maximum for one burner. They have a greater operating range, a capacity ratio of 10 to 1 being obtainable in good designs.

442. Air Registers Are Not Provided with Steam-atomizing Burners That Produce Flat Flames.—Air is usually taken in through openings in the furnace floor placed directly under the flame travel as in Fig. 331. The air rising vertically cuts across the oil and so mixes with it. This method of

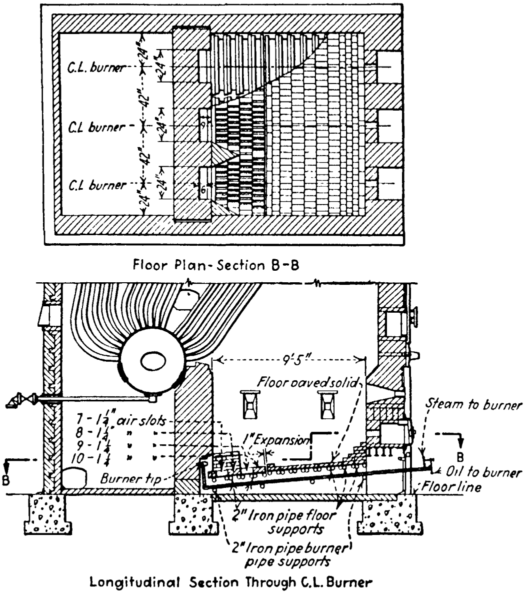


FIG. 331.—Flat-flame steam-atomizing oil burner installed under a 3,000-sq. ft. bent-tube boiler. (*Hammel Oil-Burner Equipment Company, Inc.*)

introducing air usually requires more excess air than necessary with mechanical burners.

443. In the rotary-cup type of burner, oil is atomized as it is thrown off the rim of a rotating cup. The cup may be rotated by air directed through impeller vanes as in Fig. 332 or may be motor driven (Fig. 333). In another type oil is atomized by an air-driven spinner against which the oil is directed. Burners of this type usually do not burn more than 100 to 150 gal. of oil an hour. They are frequently arranged for off and on operation for which they are well adapted.

444. Oil Burners Are Practically Always Arranged to Fire Horizontally.—In some instances the burners have been placed in the floor of large furnaces and fire vertically upward, but this arrangement is not recommended. Oil firing downward through pulverized-coal burners has been tried with

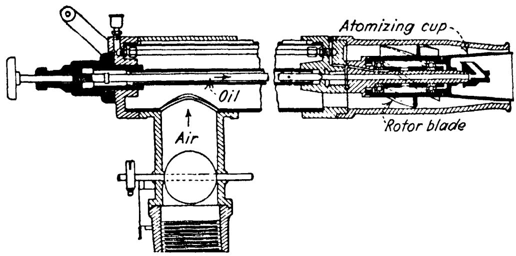


FIG. 332.—Cup-type oil burner driven by air turbine. (*Simplex Oil Heating Corporation.*)

satisfactory results. The number of burners to use in a given boiler is determined to a large extent by the width of the furnace, size of boiler, load fluctuation, and other local conditions. Considerable judgment is necessary and opinions

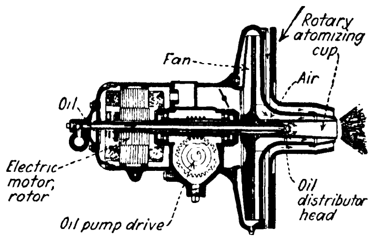


FIG. 333.—Cup-type burner with integral motor-driven fan and oil pump. (*Petroleum Heat and Power Company.*)

differ. A complete installation of a mechanical-atomizing oil burner is shown by Fig. 334.

445. Burner Selection.—Mechanical burners are generally selected for the larger installations, because of the capacities required and somewhat better economy obtainable. Steam-atomizing burners are particularly applicable where heavy refinery sludge and wastes are to be burned. They are also used to a large extent in smaller boilers and with horizontal

return-tubular boilers because with the flat and more lazy flame there is less danger of overheating the boiler shell. They cost very much less than mechanical burners, but operating cost is higher because of the use of steam for atomization. Rotary-cup type burners are occasionally used with power boilers and are particularly applicable to heating boilers operating at steam pressures less than 15 lb.

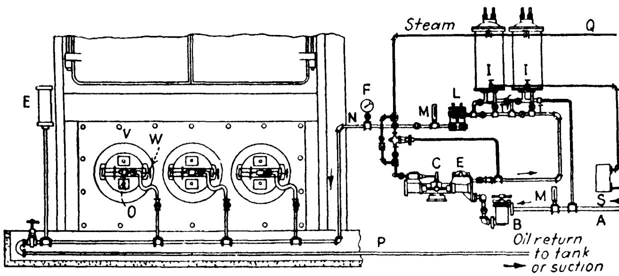


FIG. 334.—Mechanical-atomizing oil-burning system, completely installed, showing arrangement of pumps, strainers, heaters and accessories.

A Suction from oil tank
B. Duplex suction oil strainer
C. Oil pump
E. Air chamber
F. Pressure gage
I. Oil Heater
L. Duplex discharge oil strainer
M. Thermometer
N. Discharge line to burner
O. Mechanical burner
P. Return line to oil pump suction
Q. Steam line
R. Drain from heater
S. Trap
T. Relief valve on heater coils
U. Drain line to boiler-feed tank
V. Forced- or natural-draft front
W. Air control valve

446. Gaseous Fuel.—Construction of natural-gas trunk pipe lines has made this fuel available for boiler plant use in districts many hundreds of miles away from the gas fields of the South Central states, California, Ohio, New York, Pennsylvania and Virginia. The gas as it issues from the wells has its easily condensed constituents removed and is then compressed to about 400 lb. pressure and distributed over trunk lines 8 to 22 in. in diameter. Booster compressor stations at intervals of 50 to 100 miles maintain the pressure. Natural gas has a lower heating value of 900 to 1200 B.t.u. per cu. ft. It makes an ideal fuel from an operating point of view as it is clean, easily adapted to automatic control and requires but little firing equipment. Cost is its chief disadvantage. Other gases such as blast furnace and coke-

oven gas are used under boilers, but not as much as natural gas. Coke-oven gas has a heating value of about 500 B.t.u. per cu. ft. and blast furnace gas 90 to 115 B.t.u. per cu. ft.

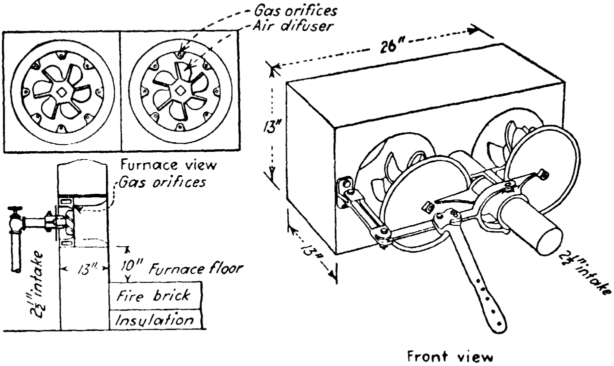


Fig. 335.—Gas burner. (*Lee B. Mettler Company.*)

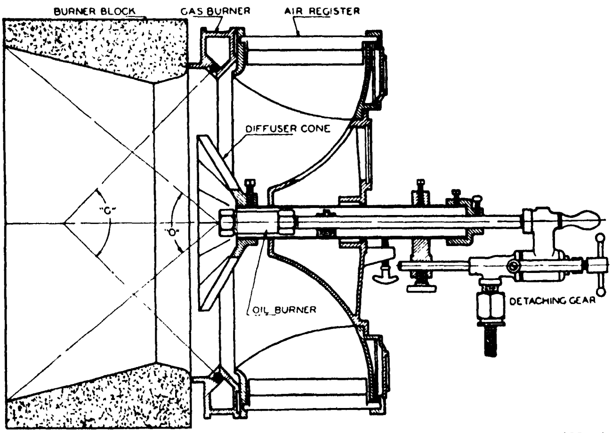


Fig. 336.—Combination gas and mechanical-atomizing oil burner. (*National Airoil Burner Company.*)

447. Gas burners are relatively simple pieces of equipment consisting of orifices or nozzles through which the gas is discharged, means for mixing the gas thoroughly with the necessary air for combustion, and controls for both air and gas quantities. Figure 335 shows one type of gas burner.

Air is drawn in by natural draft through the burner register and given a rotary motion by the deflector plates. Gas from the manifold discharges through orifices into the stream of rotating air and the turbulence so induced mixes the two thoroughly. The gas-air ratio or excess air is controlled by the register and boiler dampers. A combination oil, gas, and coal burner is shown in Fig. 336 and it will be seen the same principle of mixing gas and air is employed. In using gas great care must be taken to ventilate thoroughly the furnace and boiler setting before lighting a gas burner in order to prevent possible gas explosion which may occur if gas from a leaky valve were to accumulate.

QUESTIONS ON DIVISION 18

1. What grade of fuel oil is used most as a boiler fuel?

2. What advantages does fuel oil possess as a boiler fuel? What disadvantages?

3. What factors are important in an oil-firing installation?

4. Name three general types of oil burners.

5. At what pressure and temperature is oil supplied to mechanical oil burners?

6. How is the rate of combustion controlled with mechanical oil burners?

7. What range of control is normally possible?

8. How is oil atomized by a mechanical burner?

9. Describe one type of wide-range burner.

10. Describe a mechanical burner register.

11. What is the maximum oil quantity handled by a single mechanical burner?

12. What are two general types of steam-atomizing burners?

13. What function does steam perform in a steam-atomizing burner?

14. Describe an inside-mixing steam-atomizing burner. Describe an outside-mixing burner.

15. What is the steam consumption of steam-atomizing burners?

16. How is air supplied to a flat-flame burner? Are registers used with burners of this type?

17. May registers be used with any type of steam-atomizing burner?

18. Describe the operation of a rotary-cup burner.

19. In what direction are burners arranged to fire?

20. Describe a combination oil and gas burner.

DIVISION 19

BOILER SETTINGS AND FURNACES

448. Settings of steam boilers may be built of refractory firebrick, of a combination of firebrick insulation and common brick, or of waterwalls backed with refractory brick or insulation. Instead of solid masonry construction, suspended refractory settings may be used. Many boiler settings are completely steel encased to reduce air leakage. Settings completely enclose the boiler and furnace and only in the case of very small units is the masonry used to support the boiler. *Access and observation doors* must be provided so the operator may see the condition of his fire, and so that important parts of the boiler may be reached for inspection and repair. Access doors are usually oval in shape and are made of cast iron. They should be provided with a latch that will allow the door to open in case of pressure inside the setting.

449. Size, shape, and construction of boiler furnaces depend upon many factors, some local in character, some governed by the judgment and combustion experience of the designer. As each boiler furnace is tailor-made to fit conditions, it is not surprising that no two boiler plants have similar furnaces.

450. Furnace design is affected by the economic factors of cost, duration or hours of operation, cost of fuel, duration of peak load, and duration of minimum load; by combustion factors of efficiency, kind and character of fuel, method of firing, magnitude of peak and minimum loads, and by physical factors of space available and type of boiler. The economic end sought is minimum cost consistent with operating conditions and with life of furnace and cost of maintenance. For example, an expensive water-cooled furnace is not generally justified for a boiler operated only during the heating season and then at light loads for most of the time. In most cases

it would not be good economics to design a furnace for continuous operation at maximum peak load when peak is short and seldom occurs.

451. Many Decisions Must Be Made before Furnace Design Can Be Started.—Important among these are peak load and maximum load to be carried continuously, kind of fuel, type and character of fuel-burning equipment, and dimensions of the boiler. The furnace should be capable of sustaining the maximum continuous combustion rate with the desired excess air without slagging the furnace walls, bottom or boiler tubes. Experience indicates this can be accomplished by designing the furnace so that the ash particles entering between the boiler tubes are not above their fusion temperature.

452. Furnace volume, heat release, excess air, water-cooled surface exposed to furnace radiation, and fusion temperature of the ash are the factors in determining furnace-exit temperature. Exit-furnace temperature does not depend entirely upon heat release per cubic foot of furnace volume, nevertheless it is common practice to limit the heat released per cubic foot to approximate values depending on firing method, type of furnace, and kind of fuel. Recommended heat release values for various conditions are:

	Heat release, B.t.u. per hr. per cu. ft.	
	All-refractory furnace	Complete waterwall furnace
Pulverized coal 15 per cent CO_2 at furnace exit, Illinois coal	12,000 to 15,000	20,000 to 35,000
Eastern bituminous coal	20,000 to 22,000	25,000 to 35,000
Underfeed stoker	25,000 to 35,000	30,000 to 42,000
Chain- or traveling-grate stoker	15,000 to 25,000	30,000 to 45,000
Oil	20,000 to 30,000	30,000 to 50,000

Heat release per square foot of cold surface exposed to the furnace is a ratio used in determining furnace exit temperature. Curves such as in Fig. 337 use this ratio together with excess

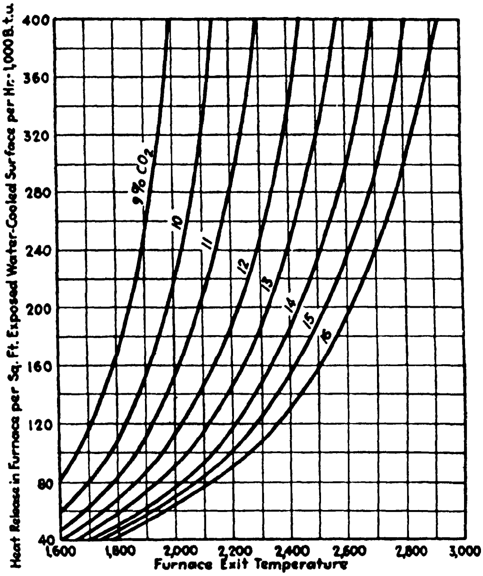
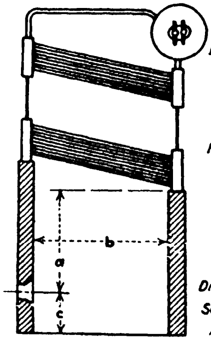


FIG. 337.—Curves for estimating gas temperature at furnace exit for various CO_2 content and heat release.



Manufacturer	Pulverized Coal X	Pulverized Coal Y	Pulverized Coal Z	Oil
Dimension c				
Flat floor, dry	4'-8" to 8'-0"	7' to 9'	5'-0" to 10'-0"	1'-6" to 2'-10"
Flat floor, wet	*	3' to 4'	3'-0" to 5'-0"	
Hopper air cooled	2'-3" to 4'-0"	5' to 6'	4'-0" to 8'-0"	
Hopper water cooled	2'-2" to 3'-9"	4' to 5'	3'-0" to 6'-0"	
Dimension a	7'-8" to 13'-0"	10' to 14'	9'-0" to 15'-0"	4' to 6'
Dimension b	10'-0" to 17'-0"	10' to 18'	11'-0" to 18'-0"	5' to 9'-6"
Distance to side wall				
Solid refractory wall	3'-2" to 5'-5"	5'-6" to 6'-6"	3'-6" to 7'-0"	2'-6" to 2'-10"
Air-cooled wall	3'-2" to 5'-5"	5' to 6'	3'-0" to 6'-6"	
Water wall	2'-10" to 5'-2"	4' to 5'	2'-6" to 6'-0"	

* Circ. burners not recommended

TABLE XV.—Minimum dimensions for locating circular pulverized-coal and oil burners of low and high capacity.

air, as indicated by CO_2 to find exit temperature, and are very helpful in furnace design.

Example.—Suppose it is desired to obtain 140,000 lb. of steam an hour from a boiler 21 ft. wide with straight tubes 24 ft. long, and that a coal having an ash-fusion temperature of 2,100° is to be burned. If an efficiency of 75 per cent is desired at full load, the heat released in the furnace will be about 194,000,000 B.t.u. per hr. With an all-refractory furnace the cold surface exposed is $21 \times 24 = 504$ sq. ft. and the heat release per sq. ft. of cold surface is 385,000 B.t.u. per hr. For satisfactory continuous operation with this coal the exit temperature should not be above 2100°. Hence from Fig. 337 it is evident that not over 10 per cent CO_2 should be carried at full load if slagging difficulties are to be avoided. This is not good operation and would not yield the efficiency desired. If, however, enough waterwall surface is installed in the furnace to bring the heat released per square foot of cold surface down to 115,000 B.t.u., full load can be carried with 13 per cent CO_2 without trouble from ash fusion, and the 75 per cent efficiency should be reached. With pulverized-coal firing, the furnace volume necessary with waterwalls would be 194,000,000 divided by say 20,000 B.t.u. per cu. ft., or 9,700 cu. ft., which would require this furnace to be 19 ft. 3 in. high, assuming the walls are vertical.

To reduce the heat release to 115,000 B.t.u. would require side and rear water walls. By adding water walls to the front walls the excess air could be further reduced and the efficiency improved or output increased without trouble from slaging. Whether this should be done depends upon whether or not the improved efficiency, together with fuel cost and operating time, will justify the additional fixed charges on the increased cost of the front wall. On the other hand, if full load will be of short duration, it is good practice to design for somewhat higher exit temperatures. Further to indicate the possible variations, the same load could be carried with high efficiency in an all-refractory furnace if a better grade of coal were used having an ash-fusion temperature not less than 2,600°. To look at it another way, waterwalls increase the capacity of a given boiler.

453. Shape of the Furnace Depends upon the Boiler and the Equipment Used for Firing.—Width of the furnace is always practically the same as width of the boiler. Depth, however, depends upon other considerations. In the case of bent-tube boilers the depth is seldom less than the projected distance between the mud drum and the front steam drum, but it may be, and often is, more than this, particularly when height of the furnace is limited by local conditions. For straight-tube boilers, depth at the furnace exit is never more

than the depth of the boiler, but may be less, as is the case when a bridge or rear-arch wall is used. For pulverized coal, underfeed stokers, oil and gas firing the tendency is to use vertical walls as much as possible. Sometimes in stokered installations it is necessary to slant the rear wall to meet the stoker dimensions.

454. With Pulverized-coal Firing, Shape Is Influenced by Direction of Firing and Method of Ash Removal.—When horizontal firing is employed, two dimensions fix the minimum height of furnace—the minimum distance from the center line of the burner to the furnace floor or start of the ash hopper, and the minimum distance from burner center line to the boiler tubes. The later dimensions must be maintained if smokeless and complete combustion is to be obtained and trouble from slag on the boiler tubes is to be avoided. These dimensions vary with the capacity of the burner and are as shown in Table XV for different makes of burner. The table also gives minimum dimensions for oil firing. Except that higher heat-release rates are permissible because of the absence of ash, oil or gas furnaces are similar in shape to furnaces fired horizontally with pulverized fuel. In the case of oil or gas, the furnace floor is always flat and usually air-cooled. With pulverized fuel, hopper bottoms are usually used only for larger installations, or when headroom is available.

455. Slag-tap Bottoms Further Influence Design of Pulverized-coal Furnaces.—Difference of opinion exists concerning the conditions favorable to the use of this method of ash disposal. Some recommend it only when the fusion temperature of the ash is not over 2250 or 2300°, while others claim successful operation is possible with coals having ash-fusion temperatures as high as 2500 to 2600°. Duration of high and low loads are other factors affecting its use. When used, some manufacturers prefer to arrange the burners either for vertical or inclined firing. Both vertical and horizontal burners are placed closer to the floor than would be the case with a dry bottom.

456. Furnace heights with underfeed stokers for high-rating large-capacity installations are 15 to 20 ft. from the floor line to the boiler header in the case of straight-tube boilers, and

10 to 14 ft. to the mud-drum center line in the case of bent-tube boilers. For small low-rating installations, setting height should not be less than about 8 ft. for straight-tube boilers and 6 ft. for bent-tube boilers. Setting heights recommended by the Stoker Manufacturers Association as minimum for smokeless operation are given in Table XVI.

457. When water cooling is applied to underfeed stokered furnaces the bridge wall or rear wall should be the first wall cooled. The next furnace cooling surface added should be in the side walls along the fuel line. Waterwall tubes at this point should be protected by cast iron or refractory blocks, both to prevent abrasion of the tubes and to provide a smooth surface to which clinker will not stick. Such side walls are usually inclined tubes parallel with the stoker and carried up about 7 tubes high. When vertical tubes are used, the entire wall is usually water-cooled and should include the ashpit. Inclined tubes sometimes make it impossible to locate the access door in the most desirable place, whereas vertical tubes do not impose this limitation. The front wall is seldom water-cooled, because the temperatures there are not so high, and in the case of straight-tube boilers the wall is not very high.

458. Chain- or traveling-grate stokers require arches; hence furnaces for them differ radically from furnaces designed for other firing equipment. Likewise, as there exists a great difference of opinion concerning the proper location and shape of these arches, the resulting variations in furnaces are almost endless. Among other functions, arches should be arranged to create turbulence, seal and protect the rear of the stoker, protect the gate, and assist ignition by directing particles of burning coal so they will fall on entering green coal.

459. Water-cooled arches permit higher ratings and placing of the arch somewhat closer to the fire. Ratings of over 400 per cent have been made possible by installing water-cooled arches. Water-cooled side walls with a cast-iron block on the tube surface decreases the maintenance of refractory along the fire line and decreases disturbance of the fuel bed caused by clinker sticking to side walls.

460. For chain-grate furnaces in which anthracite coke breeze or lignite is to be burned, the size of the front arch is

TABLE XVI.—SETTING HEIGHTS FOR VARIOUS STOKER-FIRED BOILERS RECOMMENDED BY STOKER MANUFACTURERS ASSOCIATION

	Underfeed				Chain grate				Overfeed			
	Multiple retort		Single retort		Natural draft		Forced draft*		Side feed		Front feed	
	Min.	Pref. min.	Min.	Pref. min.	Min.	Pref. min.	Min.	Pref. min.	Min.	Pref. min.	Min.	Pref. min.
Water tube:												
Horizontal, all sizes	11′ 0″	13′ 0″	9′ 0″	11′ 0″	10′ 0″	12′ 0″	12′ 0″	14′ 0″	9′ 0″	11′ 0″	9′ 0″	11′ 0″
Inclined (H.M.D.), all sizes	7′ 6″	8′ 6″	6′ 6″	8′ 6″	6′ 0″	8′ 0″	7′ 0″	8′ 0″	5′ 0″	7′ 0″	6′ 6″	8′ 0″
Inclined (V.M.D.), all sizes	6′ 0″	7′ 0″	5′ 0″	7′ 0″	4′ 0″	5′ 0″	6′ 0″	8′ 0″	3′ 6″	5′ 0″	4′ 0″	5′ 6″
Vertical (H.M.D.), all sizes	3′ 6″	5′ 0″	3′ 6″	5′ 0″	3′ 6″	4′ 6″	4′ 0″	5′ 0″	3′ 0″	4′ 0″	3′ 6″	5′ 0″
Vertical (V.M.D.), 150 hp	4′ 6″	5′ 0″	4′ 6″	5′ 0″	4′ 6″	5′ 0″	5′ 0″	5′ 6″	3′ 3″	.....	3′ 6″	4′ 6″
Vertical (V.M.D.), 250 hp	5′ 6″	6′ 0″	5′ 6″	6′ 0″	4′ 6″	5′ 0″	5′ 0″	5′ 6″	3′ 3″	.....	3′ 6″	4′ 6″
Vertical (V.M.D.), 500 hp	6′ 0″	6′ 6″	6′ 0″	6′ 6″	4′ 6″	5′ 0″	6′ 0″	6′ 6″	3′ 3″	.....	3′ 6″	4′ 6″
H.R.T.:												
72 in	8′ 0″	10′ 0″	7′ 6″	8′ 6″	7′ 0″	8′ 0″	8′ 0″	10′ 0″	7′ 0″	8′ 0″	6′ 0″	8′ 0″
84 in	8′ 0″	10′ 0″	7′ 6″	8′ 6″	7′ 0″	8′ 0″	8′ 0″	10′ 0″	7′ 0″	9′ 0″	6′ 0″	8′ 0″

Abbreviations: H.M.D. = horizontal mud drum. V.M.D. = vertical mud drum. H.R.T. = horizontal return tubular. Min. = absolute minimum. Pref. min. = preferred minimum, *i.e.*, the minimum heights recommended.

Type of Boiler	Measurement of Setting Height
Water tube:	
Horizontal	Floor line to bottom of header above stoker
Inclined (H.M.D.)	Floor line to center of mud drum
Inclined (V.M.D.)	Floor line to top of mud drum
Vertical (H.M.D.)	Floor line to center of mud drum
Vertical (V.M.D.)	Floor line to top of mud drum
Horizontal return tubular	Floor line to under side of shell.

* When burning coke breeze and anthracite fines, the setting heights indicated above should be materially increased to provide for proper arch and furnace design.

usually smaller than when bituminous coal is burned. A furnace for anthracite is shown in Fig. 338. A very short front arch is shown, its main purpose being to protect the stoker gate. The long rear arch serves several functions. At the rear end it is placed close to the gate, 9 to 10 in., so as to seal against air entering the furnace from the ashpit. Any air that may enter the furnace at this point is directed by the arch toward the front of the stoker where it will mix with and help burn the richer gas. It further shades the rear part of

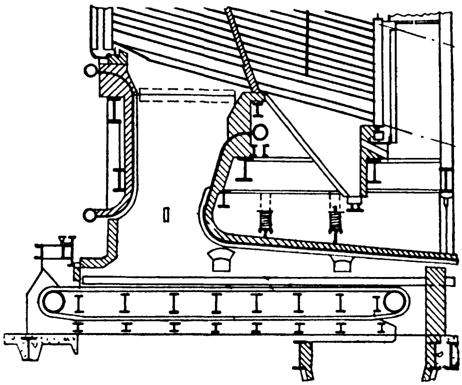


Fig. 338.—Furnace with water-cooled arches arranged for burning anthracite coal on a traveling grate stoker.

the stoker from radiation from other refractory. The arch is quite close to the stoker throughout its length and so produces a horizontal gas velocity of about 20 to 30 ft. per sec. Burning particles of coal that lift off the grate are carried forward by the high-velocity gas and deposited on the front of the stoker where they aid in igniting the entering green coal. The arch in Fig. 338 was carried far enough forward to produce a velocity of about 45 ft. per sec. at the furnace throat. Height of boiler above the grate may limit the distance the arch can be carried forward as care should be taken not to form a pocket between the arch and the first row of boiler tubes where soot may build up.

461. Front Arches Are Used When Bituminous Coal Is Fired (Fig. 339).—They are usually placed 4½ to 5 ft. above

the stoker. The distance they extend out over the stoker depends largely upon local conditions of setting height, length of stoker, and how the stoker is placed with respect to the boiler. When the main arch is set at this height, a short ignition and gate-protecting arch is usually installed under it, set 18 to 24 in. above the grate and 12 to 18 in. long. The purpose of the main arch is to throw the rich gas from the front of the stoker to the rear where it can mix with the leaner and

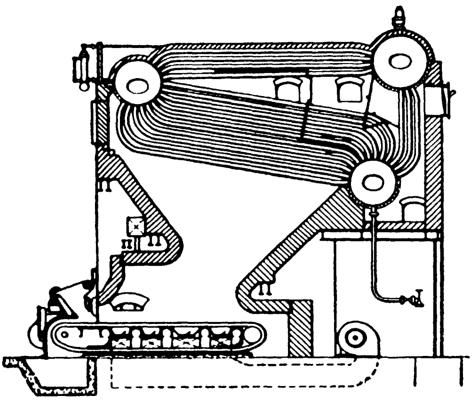


FIG. 339.—Furnace arranged for burning bituminous coal on chain grate stoker.

hotter gas from the rear of the stoker and so be completely burned.

462. Overfire air is almost always necessary for smokeless combustion with bituminous coal. It should be admitted to the furnace under the front main arch and generally not closer to the grate than 4 to 4½ ft. Overfire air supply should be taken from the main air duct ahead of the control damper, to insure ample pressure when the damper is partially closed during low loads. The rear arch for bituminous coal firing is much shorter than that used with anthracite. It is only long enough to protect the rear of the stoker and for a seal against air from the ashpit. Consequently, it is usually placed quite close to the grate.

463. Refractory service conditions are influenced by kind of fuel, volume of furnace, ratio of cooled surface to uncooled

refractory, type of firing, excess air, furnace draft, chemical composition and fusion temperature of the fuel ash, velocity of gas next to the wall or arch, height of setting, and rate of heat liberation. These various factors determine temperature of the refractory and likelihood of trouble from slag erosion or adhesion and certain types of spalling.

464. Temperature of the fire face of a refractory wall seldom exceeds 2800° and is usually less than this in underfeed-stoker installations. It cannot be said that refractory temperatures will be high in boilers operated with high-heat liberation per cubic foot of furnace volume, though in any given boiler, temperature increases with rate of heat liberation. For example, temperature measurements in two boilers, one fired with oil at a heat release of 63,000 B.t.u. per cu. ft. and the other fired with pulverized coal with a heat release of 12,000 B.t.u. per cu. ft., showed in each case a maximum refractory temperature of 2700°. Radiation from walls to cooler surfaces and flame impingement both affect refractory temperatures to a great degree. For a very high-set boiler, refractory temperatures will likely be higher than in a lower set unit because the refractory sees less water-cooled surface.

465. Refractories are subjected to very rapid temperature changes when firing up or banking a boiler. Measurements in a 13½-in. wall of a pulverized-coal-fired furnace showed a rise to 900° in 10 min. after lighting the burners and to 2100° in 1 hr. On banking, the face temperature fell from 2275 to 1950° in 10 min. Although refractories are relatively good heat conductors, nevertheless the high face temperature penetrates only a few inches. With a face temperature of 2800° the temperature 2 in. in for a 22½-in. refractory wall would be about 2500° after thermal equilibrium is established. The temperature gradient is important, for while it limits severe temperature to the face of the refractory it causes unequal expansion through the wall structure.

466. Velocity of gas in furnaces varies over a wide range. In underfeed-stoker and pulverized-coal installations, velocities range from 5 to 25 ft. a second in various parts of the furnace. Highest velocity occurs just above the fuel bed. With chain-grate stokers much higher velocities are encountered, ranging

from 10 to 60 ft. a second. Gases at the higher velocities carry with them particles of ash and erode the brickwork when it is at high temperature.

467. Characteristics of the slag formed from the ash in the fuel burned are the principal factor governing the maximum allowable temperature in a furnace. For each combination of slag and refractory there is a temperature above which brick walls are rapidly eroded. Some engineers fix this critical temperature at the softening temperature of the slag.

468. Softening temperature of coal ash varies from 2000 to 3000°. Coal-ash slags are classed as acid slags, and their

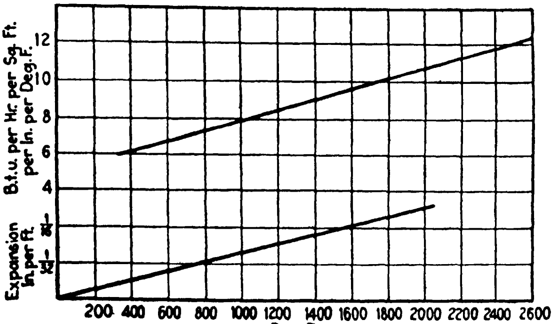


FIG. 340.—Thermal expansion and conductivity of average fireclay.

softening temperatures are usually lower than that of the ash from which they are formed. Coal-ash slags and refractories contain similar materials, and because of this, softened or fluid slag readily adheres to the wall and may enter into combination with the brickwork.

469. Slags are of two general types.—Those that adhere to the wall, forming a coating of increasing thickness, and those that flow over the wall, leaving only a thin coating. Slags of the first type damage the refractory only when they are removed during cleaning. The flowing slag of the second type usually combines with the refractory and corrodes or erodes the wall.

470. Refractory materials used in boiler furnaces to meet the service conditions outlined are fire clay, kaolin, diaspore, cyanite, silicon carbide, and chrome.

471. First-quality firebrick are made of fire clay having an alumina content of about 40 per cent and fusion temperature of 3000 to 3100°. Second- and third-quality firebrick have lower alumina content and lower fusion temperature and should never be used in lining combustion chambers.

472. Kaolin is used in the manufacture of superrefractory brick that are very dense, have an alumina content of 43 to 45 per cent and a fusion temperature of 3100 to 3250°. To avoid volume changes in service, brick of this material have a high percentage of precalined kaolin and are burned at higher temperature than fire-clay brick.

473. Cyanite and diaspore are used mixed with fire clay to give special refractories of high alumina content. By varying the amount of fire clay used, the alumina content in the resulting brick can be governed. Special brick containing 50, 60, 70, and 80 per cent alumina are standard products. The fusion temperatures for such brick are, respectively, 3200 to 3250, 3290 to 3300, 3335, and 3389°. Early high-alumina brick gave trouble from shrinkage, but recent developments have eliminated this difficulty.

474. Diaspore is used almost entirely for these special refractories, because cyanite has to be imported from India, hence is too expensive. This type of refractory has given some trouble from spalling, but in the two lower grades the spalling tendency has been greatly reduced, and in fact some of the diaspore refractories now show high resistance to spalling. They have excellent resistance to the slagging action of ash of high basic oxide content but are fluxed by coal slags having a high percentage of lime. High-alumina brick are practically inert to the influence of CO, and SO_2 weakens them less than it does fire-clay brick.

475. Silicon Carbide Is a Product of the Electric Furnace.—It is a compound of silicon and carbon produced at a temperature in excess of 4000°. It is formed into refractory brick and shapes by bonding selected grain sizes with small percentages of refractory bonds. Characteristics of brick made of this refractory are strength at high temperature, density, resistance to abrasion, infusibility, low spalling, uniformity of size, high thermal conductivity—the latter quality making

this refractory suitable as a facing material on water-cooled walls. It maintains practically its full crushing strength to 2462° and at a temperature of 3092° shows no evidence of softening under a compressive load of 175 lb. per sq. in. In reducing atmospheres, silicon carbide refractory is practically unaffected by slags, and for this reason is often used at the clinker line in stoker-fired furnaces. In an oxidizing atmosphere, as occasionally occurs in the upper portion of a furnace, oxidation begins at temperatures as low as 1550°, causing gradual growth or permanent expansion. Likewise, under this condition it is decomposed at ordinary furnace temperatures by slags high in iron. If air is present, silicon carbide should be cooled to take advantage of its slag resistance.

476. Chrome is being used for high-temperature cements and as a plastic refractory on waterwalls and furnace bottoms. It is not used in boiler furnaces in brick form because of its high spalling characteristics. It is the most neutral refractory known, hence is highly resistant to slag erosion. Its fusion temperature ranges between 3400 and 3700°.

477. Plastic fire clay refractories have a considerable field of usefulness for repairing boiler settings, especially around doors, burners, and other places where special shapes are used. They are also used for completely lining a furnace, giving a monolithic structure. To get the most satisfactory results from plastic refractory used as a furnace lining, it is essential that it be installed by masons who have had extensive experience with its application. Likewise provision should be made for expansion by building the wall in panels.

478. When Selecting Refractories, Fusion Temperature Is of Secondary Importance.—This temperature is never reached in boiler furnaces, and first-quality brick rarely fail from actual fusion unaided by fluxing with slag. Of much greater importance is the ability of the refractory to stand load at high temperature. Load-carrying ability of firebrick is influenced by the quality of the fire clay, density of the brick, and temperature at which it is burned in the kiln. Dense hard-burned brick usually have greater strength than softer burned brick containing a high percentage of flint clay. Load tests are made by subjecting the brick to 25 lb. per sq. in., raising the

temperature, and noting per cent deformation. Fire-clay brick subjected to this test usually expand up to a certain temperature and then deform rapidly as shown by the curves in Fig. 341.

479. Chemical analysis of brick tells the engineer practically nothing about its characteristics, particularly with respect to its fluxing action with fuel-ash slag and its consequent ability to withstand slag erosion. The damaging action is not only influenced by the respective compositions of the refractory and ash but also by the temperature and furnace atmosphere

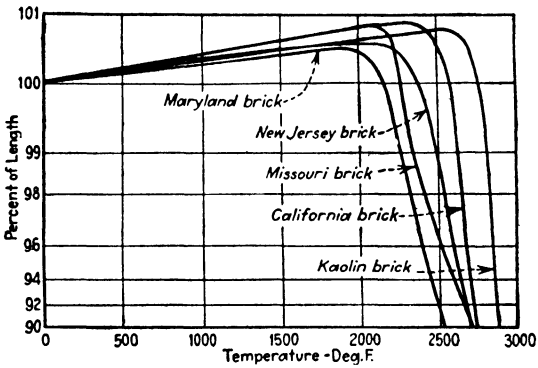


FIG. 341.—Deformation of fireclay bricks under load of 25 lb. per sq. in. at various temperatures.

in contact with the wall. If this is high in CO and low in O_2 the fluxing temperature is generally much lower. Generally speaking, a dense brick withstands slag action better than a porous brick. Coating the walls with chrome mortar and using chrome in laying up the brick has in some instances proved beneficial. Chrome and silicon carbide withstand slag better than any other boiler refractory.

480. Spalling, which is the breaking away of solid pieces from the fire side or end of the brick, may be due to one or more of several causes. One cause of such wastage is the volume change caused by the rapid temperature fluctuation occurring when firing up or banking a boiler. Usually porous or less dense brick of soft burn give less trouble with spalling from this cause than do dense brick. The porous structure

distributes the stresses and permits a large portion of the expansion to be taken up within the brick. Lack of uniformity in the refractory material may cause a porous brick to spall more than a denser brick.

481. When slag forms on a brick it sometimes penetrates into the brick a considerable distance, forming a very brittle section which may easily rupture or spall when subjected to any strain. The more porous the brick, the more readily it is penetrated by molten slag and the greater the tendency to spall from this cause. Another cause of spalling is pinching of the fire ends of the brick when insufficient clearance for expansion has been allowed. It should be borne in mind that the temperature of the fire end of the brick is materially higher than the rest of the brick and that it consequently expands more than the rest of the brick. If the brick does not reach the softening temperature before the load becomes too great, relief must be obtained by failure in such a way as to relieve the strain. Consequently, other factors being equal, brick which have the highest expansion and least uniform rate of attaining their maximum expansion are most susceptible to spalling when subjected to rapid heating and cooling.

482. There Is No "Cure-all" or "Best" Refractory.—To resist certain furnace conditions, a dense brick may be best, while for other conditions a more porous brick is indicated, and both conditions may exist in the same furnace at the same time. Where conditions are severe, as in oil furnaces, brick with higher fusion temperature and load-bearing characteristics may be required, but in all cases the brick resulting in the lowest total cost, including maintenance, should be used.

483. In settings of suspended-wall construction (Figs. 342, 343, and 344) the refractory is built in sections and each is supported by brackets or castings which in turn are fastened to steel beams or columns. Each tile is held by the castings from falling into the furnace but free to move vertically. Sections vary from 2 to 11 brick high depending upon severity of furnace conditions. Depth of refractory tile between furnace and face of castings varies from 8½ to 12 in. A space left between each section takes care of vertical expansion. Offset tile are usually used at these expansion joints which are

packed with asbestos or mineral wool. Distance between hangar castings and dimensions of the tile between adjacent

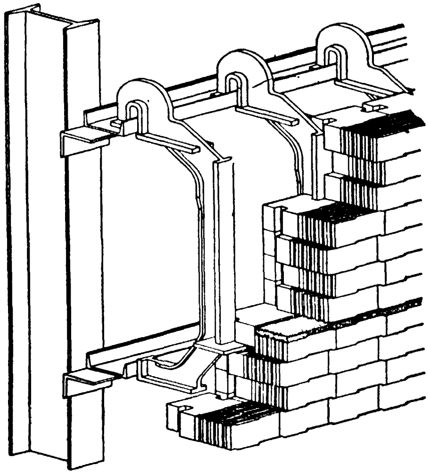


FIG. 342.—Detrick air-cooled panel construction.

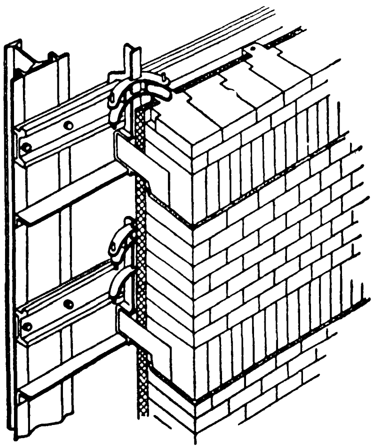


FIG. 343.—American Arch insulated wall, panel construction.

castings are proportioned so that space is provided for horizontal expansion of each tile individually. Tile of suspended

walls are usually set in fire clay making repair easier and providing for expansion as the clay shrinks with the first fire. When expansion joints in the horizontal direction are provided, air setting high temperature cement may be used.

484. An air-cooled setting may be made with suspended-wall construction by leaving a space of 4 to 6 in. between the refractory tile and an outside casing of steel and insulation. Air circulated in a setting of this type becomes heated and is used in the furnace to support combustion.

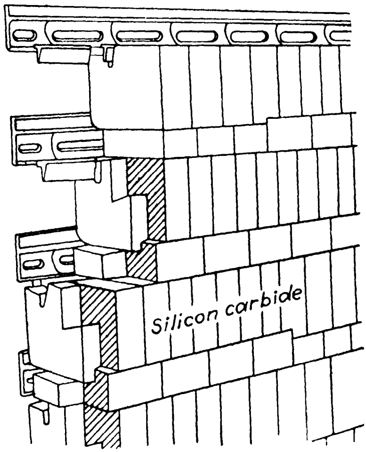


FIG. 344.—Bigelow-Liptak wall faced with silicon-carbide refractory.

485. Suspended refractory walls offer several advantages over solid refractory construction. Although first cost is higher, cost of repairs are lower because less refractory has to be replaced and the work can be done more easily and quickly. Only the section that has eroded or spalled need be removed and replaced with new tile and the rest of the wall left untouched. The refractory carries less load; hence it can operate at a higher temperature. More positive means for taking care of expansion are provided.

486. Arches used with chain grates are always of the suspended type similar in many respects to suspended walls. Several types of arch construction are shown in Figs. 345,

346, and 347. The upper or cool side of an arch should never be closed in unless special provision is made to ventilate it

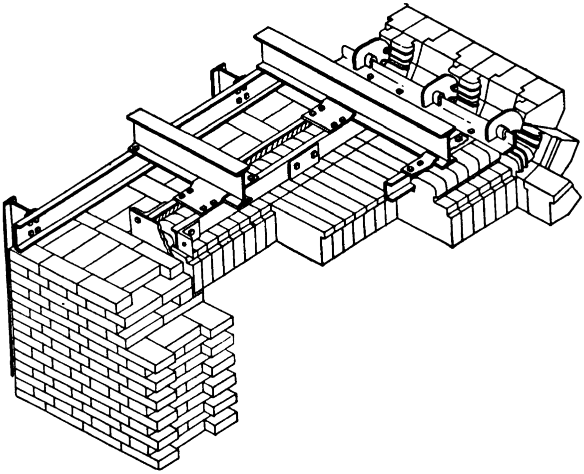


FIG. 345.—Arch in which alternate rows of tile are removable without disturbing channel suspension. (*American Arch.*)

as otherwise some of the supporting steel may be overheated. Arches are always made up with fire clay to facilitate repair

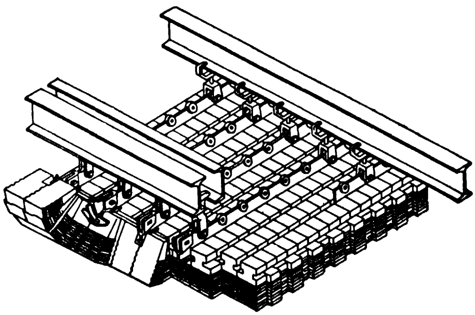


FIG. 346.—Arch with Detrick tile corrugated on all sides to prevent spalled refractory from falling into the furnace.

and provide for expansion. A feature of the suspended-type arch is the ability to remove broken tile and replace with new without destroying or removing more than a few tile.

487. When waterwalls are used the boiler setting is as illustrated in Div. 7 on Waterwalls. Parts of the boiler not enclosed by waterwalls are enclosed either with solid refractory or suspended refractory walls.

488. Boiler settings should be constructed only by masons who have had extensive experience with this kind of work, because conditions existing in a furnace wall are so different from those in other masonry. From the point of view of the

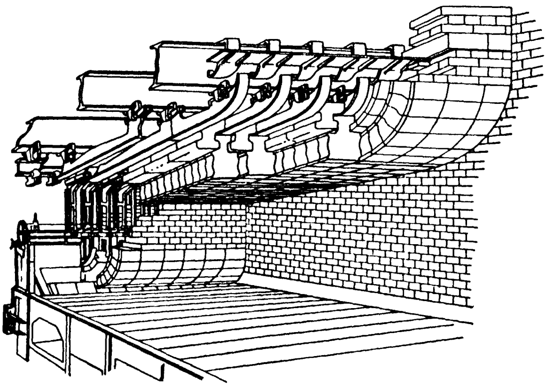


FIG. 347.—Bigelow-Liptak, double-suspension arch increases depth of refractory that can be eroded or spalled.

mason, the most important of these special factors are expansion (which is different in different parts of the setting and also throughout the wall thickness), airtight construction, and variation in brick size from specified dimensions. Brick variations over 2 per cent, plus or minus, from specified length, width, and thickness should not be tolerated.[1]

489. When laying up firebrick, experience indicates that best results are obtained when brick are placed as close together as possible. One course should be laid at a time and care should be taken to see that it is level. This may be done either by rejecting brick thinner or thicker than those already laid or by rubbing each course with silicon carbide brick or other abrasive until a substantially plane and level surface is obtained. Either fire clay or high-temperature

[1] See "Standards of A.S.T.M." on "Refractories."

air-setting mortar may be used as a bonding material. Those who advocate fire clay claim each brick has a better opportunity to expand and that there is less likelihood of chemical reaction between brick and bonding material when the fire clay used is similar to that used in the brick. Those who advocate high-temperature mortars claim a more monolithiclike structure of greater strength.

490. When fire-clay bond is used, a good mix consists of 60 per cent calcined clay and about 40 per cent raw clay, all passing through a 40-mesh sieve. Water should be added until the mixture has the consistency of thick cream so that the joint is less than $\frac{1}{16}$ in. thick. Each brick should be dipped into the fire-clay mix and then pushed sideways into place so as to work the bond into irregularities in the face of the brick. When the brick is in place it should be tapped with a hammer to make sure the joint is as thin as possible. By laying brick in this manner, it is seldom necessary to use a dipper to pour fire-clay batter over the brick. "Buttering" of brick is not good practice. When high-temperature mortar bond is used the procedure is the same, except that with chrome mortar slightly thicker joints are permissible.

491. In Laying Up Brick Be Sure to Stagger All Joints.—Vertical joints should not come one above another, and joints extending straight through the wall should be avoided. Voids in the wall filled with fire clay should not be permitted. They prevent a proper bond in the center of the wall where it is most needed.

492. Transverse Bonding of the Wall Is Important for Structural Strength.—The strongest transverse bond is the Old English bond (Fig. 348) of alternate header and stretcher courses. But this arrangement makes it hard to replace the firebrick lining and is little used in furnace brickwork. In places where replacements are not likely, such as the upper part of the setting behind boiler tubes, this bond can be used to advantage. A bond construction that permits easier replacement is shown in Fig. 349. This is a running bond for a wall having 9 in. of first-quality brick lining and 9 in. of second-quality brick. First-quality brick are laid three header sources and a stretcher, and the second-quality bricks

are laid three stretchers and a header. Note transverse bonding at every fourth course by headers stepping down

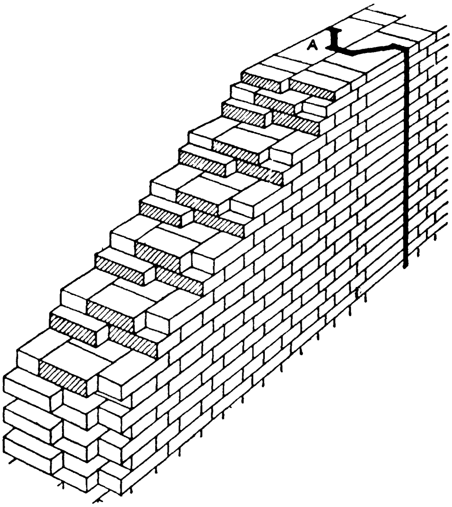


FIG. 348.—English bond makes the strongest wall. Special shapes for an expansion joint are shown at *A*.

from inside to outside of wall. Also note that all vertical joints are staggered. Walls with four header and a stretcher are also used a great deal.

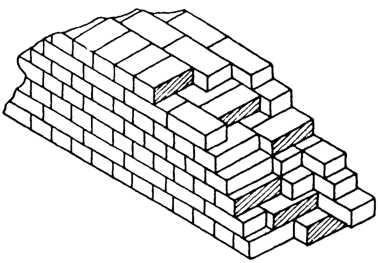


FIG. 349.—Running bond with three header to each stretcher coarse.

493. Wall Thickness and Composition.—For structural stability a wall $13\frac{1}{2}$ in. thick should not be higher than 10 ft., an 18-in. wall may be built up to 26 ft. in height, and a 22-in.

wall may be built to 35 ft., if of first-quality brick, and to 40 ft. if of super or special refractory. Furnace linings should never be less than 9 in. Many engineers prefer walls of first-quality refractory throughout their thickness, but for thick walls this seems rather wasteful. Red or common brick for the outer part of medium high walls have been used successfully, but for very high walls such usage is not con-

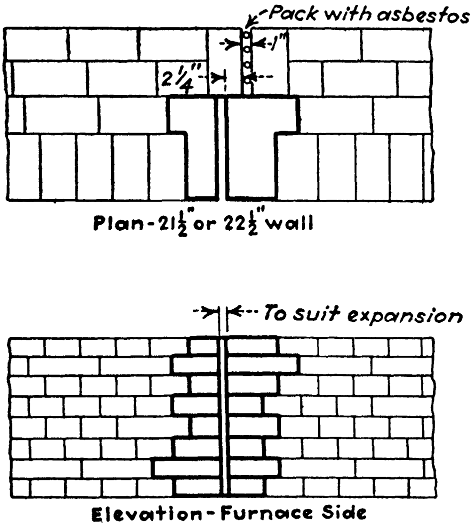


FIG. 350.—Expansion joints should be vertical in elevation and staggered in plan to reduce air leakage.

sidered good practice as differential expansion is often enough to break the bond between the two parts of the wall and so disrupt the structure. This also applies to insulating brick; they should never be tied into the firebrick part of the wall. When insulation is used it should be supported separately from the brick of the setting.

494. Expansion in a boiler setting takes place in three directions—vertically, horizontally along the plane of the wall, and at right angles to it. Vertical expansion is always upward and is usually easy to provide for by allowing the wall to move upward freely without coming in contact with

steel work or being hindered by beams embedded in the wall and tied to building steel. Sometimes vertical expansion gives trouble, as, for example, when one end of a side wall is exposed to the furnace while the other end is behind boiler

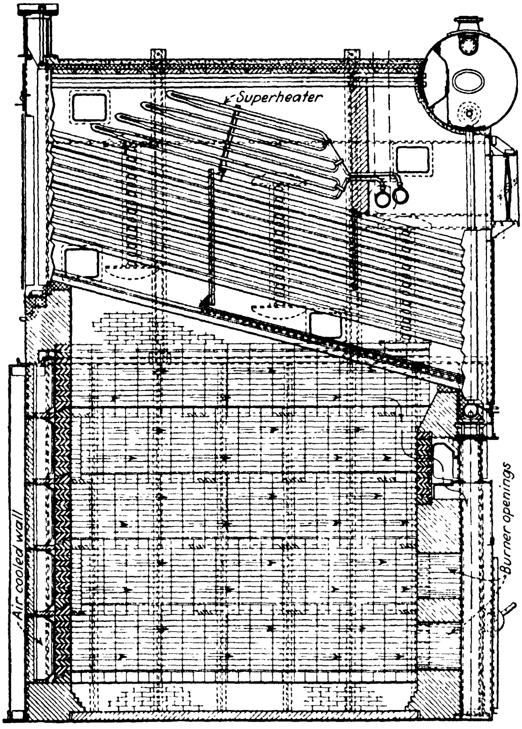


FIG. 351.—Furnace for pulverized coal. Air-cooled suspended-wall refractory setting for 7,630 sq. ft. Springfield boiler.

tubes. The higher temperature of the part exposed to the furnace makes this portion of the wall expand more and so tends to lift the cooler part with it, often causing a large diagonal crack in the cooler part of the wall. When this condition exists, trouble can be avoided by building a vertical expansion joint as nearly as possible at the dividing temperature zones.

495. A Furnace Wall Expands as It Heats Up About $\frac{1}{16}$ In. per Foot of Length.—If the wall is long, large forces are required to push great masses of masonry bodily toward an expansion joint. Experience has shown expansion joints should be placed every 5 or 6 ft. to avoid forces that might crush the brick or buckle the wall. Figures 348 and 350

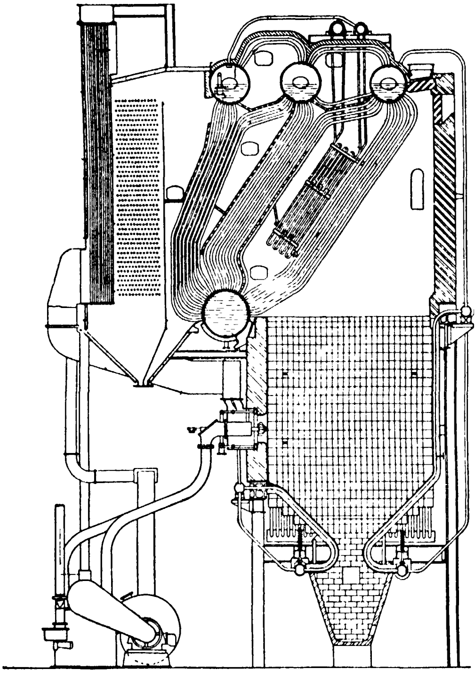


FIG. 352.—Pulverized-coal-fired furnace with three water-cooled walls protected by cast-iron blocks. Upper part of setting and front wall are of solid refractory.

show how such joints are constructed. The space left in the joint should be 1 to $1\frac{1}{2}$ in. if high-temperature air-set mortar is used and $\frac{3}{4}$ to $1\frac{1}{4}$ in. if firebrick are set in fire clay. Expansion joints should be packed loosely with asbestos or mineral wool to stop air leakage. The action of slag sometimes pulls the brick loose at these joints but special-shaped interlocking tile as in Fig. 350 are available to prevent this. Expan-

sion joints should extend vertically up through the height of the wall. It is the usual practice to extend bridge walls

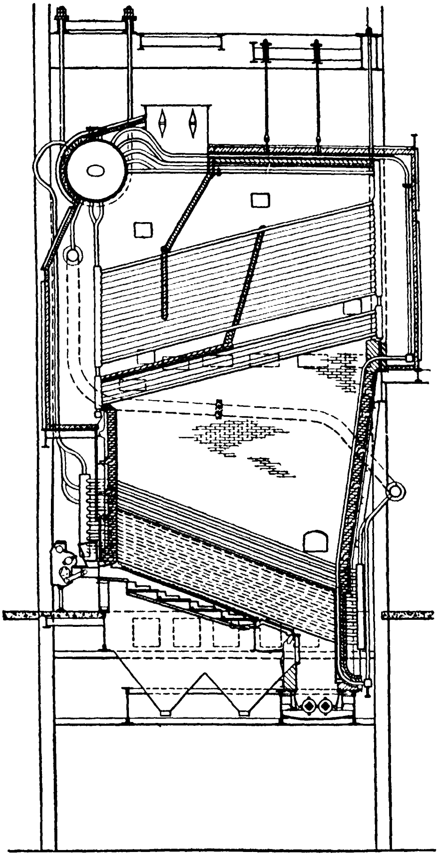
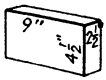


FIG. 353.—Furnace for multiple-retort stoker-fired boiler with water walls backed with refractory.

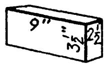
into the side walls, leaving space at the ends for horizontal expansion.

496. Buck stays are vertical steel members used to take the thrust when an arch is used in the boiler setting and to strengthen the wall. When possible they should be placed

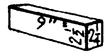
37 STANDARD 9-IN. FIREBRICK SHAPES



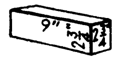
9-IN. STRAIGHT
$9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ in.



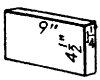
SMALL 9-IN. BRICK
$9 \times 3\frac{1}{2} \times 2\frac{1}{2}$ in.



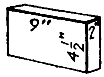
SOAP
$9 \times 2\frac{1}{2} \times 2\frac{1}{4}$ in.



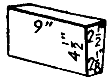
CHECKER
$9 \times 2\frac{3}{4} \times 2\frac{3}{4}$ in.



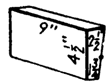
SPLIT BRICK
$9 \times 4\frac{1}{2} \times 1\frac{1}{4}$ in.



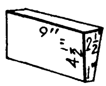
2-IN. BRICK
$9 \times 4\frac{1}{2} \times 2$ in.



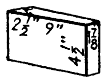
NO. 1 ARCH
$9 \times 4\frac{1}{2} \times (2\frac{1}{2}-2\frac{1}{8})$ in.
76 brick to the circle
$4\frac{1}{4}$ ft. I.D.; 5 ft. O.D.



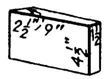
NO. 2 ARCH
$9 \times 4\frac{1}{2} \times (2\frac{1}{2}-1\frac{3}{4})$ in.
38 brick to the circle
$1\frac{3}{4}$ ft. I.D.; $2\frac{1}{2}$ ft. O.D.



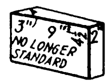
NO. 3 ARCH
$9 \times 4\frac{1}{2} \times (2\frac{1}{2}-1)$ in.
19 brick to the circle
6 in. I.D.; $1\frac{1}{4}$ ft. O.D.



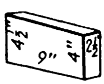
NO. 1 WEDGE
$9 \times 4\frac{1}{2} \times (2\frac{1}{2}-1\frac{7}{8})$ in.
91 brick to the circle
$4\frac{1}{2}$ ft I.D.; 6 ft. O.D.



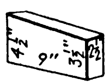
NO. 2 WEDGE
$9 \times 4\frac{1}{2} \times (2\frac{1}{2}-1\frac{1}{2})$ in.
57 brick to the circle
$2\frac{1}{4}$ ft. I.D.; $3\frac{3}{4}$ ft. O.D.



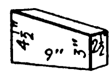
NO. 3 WEDGE
$9 \times 4\frac{1}{2} \times (3-2)$ in.
57 brick to the circle
3 ft I.D.; $4\frac{1}{2}$ ft. O.D.



NO. 1 KEY
$9 \times (4\frac{1}{2}-4) \times 2\frac{1}{2}$ in.
113 brick to the circle
12 ft. I.D.; $13\frac{1}{2}$ ft. O.D.



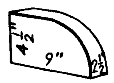
NO. 2 KEY
$9 \times (4\frac{1}{2}-3\frac{1}{2}) \times 2\frac{1}{2}$ in.
57 brick to the circle
$5\frac{1}{4}$ ft. I.D.; $6\frac{3}{4}$ ft. O.D.



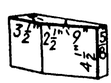
NO. 3 KEY
$9 \times (4\frac{1}{2}-3) \times 2\frac{1}{2}$ in.
38 brick to the circle
3 ft. I.D.; $4\frac{1}{2}$ ft. O.D.

NO. 4 KEY
$9 \times (4\frac{1}{2}-2\frac{1}{4}) \times 2\frac{1}{2}$ in.
25 brick to the circle
$1\frac{1}{2}$ ft. I.D.; 3 ft. O.D.

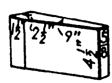
FEATHER EDGE
$9 \times 4\frac{1}{2} \times (2\frac{1}{2}-\frac{1}{8})$ in.



JAMB BRICK
$9 \times 4\frac{1}{2} \times 2\frac{1}{2}$ in.

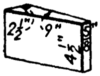


NO. 1 NECK
$9 \times 4\frac{1}{2} \times 2\frac{1}{2} \times 3\frac{1}{2} \times \frac{5}{8}$ in.



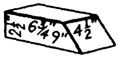
NO. 2 NECK
$9 \times 4\frac{1}{2} \times 2\frac{1}{2} \times 1\frac{1}{2} \times \frac{5}{8}$ in.

37 STANDARD 9-IN. FIREBRICK SHAPES (CONTINUED)



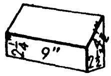
NO. 3 NECK

$9 x 4\frac{1}{2} x (2\frac{1}{2}-\frac{5}{8})$ in.



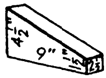
END SKEW

$(9-6\frac{3}{4}) x 4\frac{1}{2} x 2\frac{1}{2}$ in.



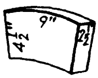
SIDE SKEW

$9 x (4\frac{1}{2}-2\frac{1}{4}) x 2\frac{1}{2}$ in.



EDGE SKEW

$9 x (4\frac{1}{2}-1\frac{1}{2}) x 2\frac{1}{2}$ in.



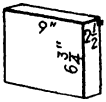
CIRCLE BRICK

$9 x 2\frac{1}{2} x 4\frac{1}{2}$ in.

12 to 45 brick to the circle

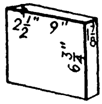
24 to 120 in. I.D.;

33 to 129 in. O.D.



LARGE 9-IN. STRAIGHT

$9 x 6\frac{3}{4} x 2\frac{1}{2}$ in.

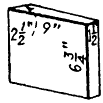


LARGE 9-IN. NO. 1 WEDGE

$9 x 6\frac{3}{4} x (2\frac{1}{2}-1\frac{7}{8})$ in.

91 brick to the circle

$4\frac{1}{2}$ ft. I.D.; 6 ft. O.D.

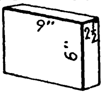


LARGE 9-IN. NO. 2 WEDGE

$9 x 6\frac{3}{4} x (2\frac{1}{2}-1\frac{1}{2})$ in.

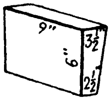
57 brick to the circle

$2\frac{1}{4}$ ft. I.D.; $3\frac{3}{4}$ ft. O.D.



FLAT BACK STRAIGHT

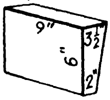
$9 x 6 x 2\frac{1}{2}$ in.



NO. 1 FLAT BACK ARCH

$9 x 6 x (3\frac{1}{2}-2\frac{1}{2})$ in.

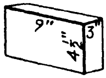
$2\frac{1}{2}$ ft. I.D.; $3\frac{1}{2}$ ft. O.D.



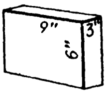
NO. 2 FLAT BACH ARCH

$9 x 6 x (3\frac{1}{2}-2)$ in.

$1\frac{1}{3}$ ft. I.D.; $2\frac{1}{3}$ ft. O.D.

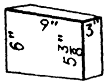


$9 X 4\frac{1}{2} X 3$ IN. STRAIGHT



9X6X3-IN. STRAIGHT

Also $9 x 6 x 2\frac{1}{2}$ in.



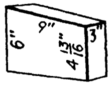
9x6x3 IN. NO. 1 KEY

$9 x (6-5\frac{3}{8}) x 3$ in.

91 brick to the circle

12 ft. 11 in. I.D.;

14 ft. 5 in. O.D.



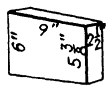
9X6X3 NO. 2 KEY

$9 x (6-4\frac{13}{16}) x 3$ in.

48 brick to the circle

6 ft. 1 in. I.D.;

7 ft. 7 in. O.D.



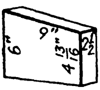
$9 X 6 X 2\frac{1}{2}$ IN. NO. 1 KEY

$9 x (6-5\frac{3}{8}) x 2\frac{1}{2}$ in.

91 brick to the circle

12 ft. 11 in. I.D.;

14 ft. 5 in. O.D.

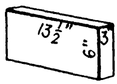


$9 X 6 X 2\frac{1}{2}$ IN. NO. 2 KEY

$9 x (6-4\frac{13}{16}) x 2\frac{1}{2}$ in.

48 brick to the circle

6 ft. 1 in. I.D.; 7 ft. 7 in. O.D.

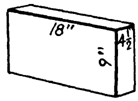


$13\frac{1}{2}$ IN. STRAIGHT

$13\frac{1}{2} x 6 x 3$ in.

Also $13\frac{1}{2} x 6 x 2\frac{1}{2}$ in.

TWO SPECIAL SIZES



BOTTOM BLOCK

$18 X 9 X 4\frac{1}{2}$ IN.

in back of expansion joints. They should be in place before the wall is built so it may be built to the buckstay, thus obtaining a firm bearing over its full length. All steelwork should, of course, be outside of walls of masonry type. When for any reason this becomes impossible, space should be provided for air to circulate and for expansion of the setting.

QUESTIONS ON DIVISION 19

1. Of what are boiler settings built?

2. What factors affect furnace design?

3. What determines furnace-exit temperature?

4. What determines furnace volume?

5. What is the meaning of "heat release per square foot of cold surface"? How is this ratio used in furnace design?

6. Upon what does the shape of the furnace depend?

7. How close may horizontal coal burners be placed to the furnace side walls? How close to the furnace bottom?

8. What height of furnace should be allowed with underfeed stokers of large capacity and high rating?

9. When water cooling is applied to furnace walls which wall should be cooled first?

10. What is the purpose of arches with chain-grate stokers?

11. Describe the length and location of arches when bituminous coal is burnt. When anthracite coal is burnt.

12. When is overfire air required?

13. What is the highest temperature imposed on furnace refractory?

14. Does refractory temperature depend on heat release per cubic foot of furnace volume?

15. What are two different kinds of slags encountered in boiler furnaces?

16. What refractory materials are used in boiler furnaces?

17. About what is the alumina content of first-quality firebrick? What is their fusion temperature?

18. What are the characteristics of brick made of kaolin?

19. What are the characteristics of brick made with mixtures of fire clay and diaspore?

20. What is silicon carbide? What are the characteristics of brick made with silicon carbide?

21. Why is not chrome ore used to make firebrick for use in boilers?

22. How important is fusion temperature of the brick when selecting boiler refractory?

23. What influences the ability of a brick to carry load at high temperature?

24. Define spalling.

25. What are three causes of spalling?

26. Describe one type of suspended-wall construction.

27. What are the advantages of suspended-wall construction?

28. How are arches supported?

29. Describe how firebrick should be laid up.

30. What is a good mix for fire-clay bond? How should it be applied to the brick?

31. Describe the old English bond. Describe a running bond.

32. To what maximum height may a refractory wall 13½ in. thick be built? How high may a 22-in. wall be built?

33. How does expansion take place in a boiler setting?

34. How much does a refractory wall expand?

35. Describe the construction of an expansion joint.

36. How much space should be left in expansion joints?

37. How far apart should expansion joints be placed?

DIVISION 20

DRAFT AND ITS PRODUCTION AND MEASUREMENT

497. The function of draft is to force air to the fire and to carry away the gaseous products of combustion. Proper combustion in a furnace can occur only when an ample quantity of oxygen, which is contained in the air, is supplied to the burning fuel. If the supply of oxygen is insufficient, the combustion will be sluggish and inefficient—even with superrefined furnace construction and the most skillful stoking.

498. Chimney draft is the pressure difference indicated in Figs. 354 and 355. The impression that it is due to a vacuum is erroneous. Why this is true will be explained later. A draft-gage reading (Fig. 356) indicates the tendency or pressure of the cold outside air to force its way into the furnace, boiler setting, breeching, or chimney.

499. Draft Pressure May Be Produced either Naturally or Artifically.—Only chimney, or *natural draft* is discussed in this division. Mechanical *draft* is discussed in Div. 22. A classification may be arranged thus:

1. Natural draft.
   A. Produced by the natural draft of a chimney.
2. Artificial draft.
   A. Steam jet.
      a. Induced.
      b. Forced.
   B. Mechanical draft by a fan.
      a. Induced.
      b. Forced.

500. The function of a chimney is twofold: (1) It produces the draft, whereby the air and gas are forced through the furnace, fuel, boiler, and setting. The air which carries the oxygen, necessary for the proper combustion—burning—of the fuel, is thereby furnished to the fuel bed. (2) It carries the products of combustion to such a height before discharg-

ing them that they will not be objectionable or injurious to surroundings.

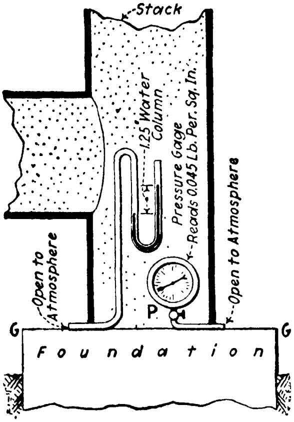


FIG. 354.—Pressure gages inside stack.

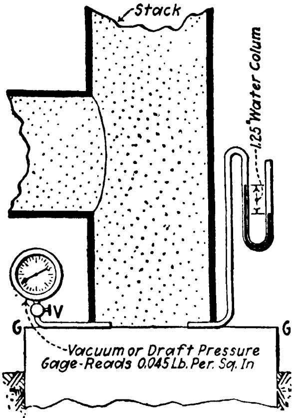


FIG. 355.—Vacuum gage outside stack.

501. The basic principle of natural or chimney draft is as follows: When a lighter gas is submerged in a heavier one, the lighter gas is forced upward by the heavier. A hot-air balloon ascends in the cooler atmosphere; similarly a cork, which is submerged in water, rises. It is due to the same fundamental causes that a chimney produces the pressure difference which forces the gas through the boiler furnace and setting.

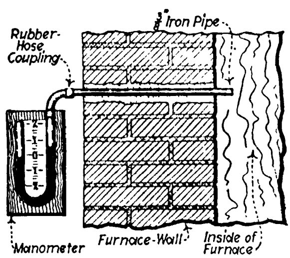


FIG. 356.—Manometer draft gage arranged for indicating difference in pressure between inside and outside of boiler setting.

Example.—A cubic foot of air at a temperature of 60°F. weighs about 0.08 lb. A cubic foot of air at 600°F., weighs about 0.04 lb. Hence a cubic foot of 600° air which is submerged in an atmosphere of 60° air, will be forced upward with a force of 0.08 − 0.04 = 0.04 lb.

502. How a chimney produces a draft may be understood from a consideration of Figs. 357 to 359.

Explanation.—In Fig. 357, stacks C and H are of identical dimensions and construction. CG is filled with the cool air of the surrounding atmosphere. This CG air is at the same temperature as that of the surrounding atmosphere. HF is filled with hot air. The volume of hot air in H, because of its lower density (see the author's "Practical Heat"), weighs less than the equal volume of cool air in C. The equal cool-air columns A_1 and A_2, each extending vertically upward to the limit of the atmosphere counterbalance one another. Hence (see preceding section) the heavy volume C tends to force the lighter volume H up out of the stack.

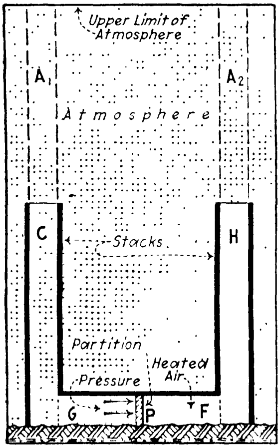


Fig. 357.—Illustrating the principle of chimney draft.

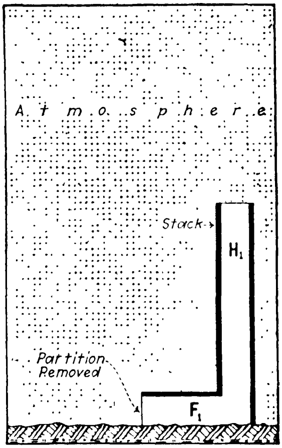


Fig. 358.—Furnace and stack.

Thus a pressure is exerted against partition P. The force, in pounds, imposed against P = (lb. weight of C) − (lb. weight of H).

Neither the volume of hot air in the horizontal passage F nor that of cool air in G are factors in the problem since neither increases the *vertical* heights of the air columns.

If CG were removed (Fig. 358) and the entering cool air were continuously heated in F_1, a continuous current of air would, obviously, be forced upward through F_1H_1. The removal of stack CG has not, since CG contained only air at atmospheric temperature, affected the elements of the situation. Figure 359 illustrates the application of the principle to an actual boiler and stack. In Fig. 359, the pounds draft pressure impressed against the grate = (lb. weight of cool-air column D) − (lb. weight of hot air column B). D and B both have the same cross-sectional area. The height of columns B and D is the vertical distance between the top of the stack and the bottom of the grate.

503. Draft pressure is measured in "inches of water column" in practice because this is the most convenient unit.

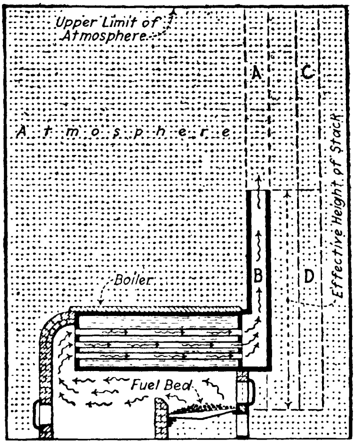


FIG. 359.—Draft-pressure principles applied to an actual boiler.

It could, if desirable or convenient, be measured in any unit of *pressure per unit area* such as "pounds per square inch."

Example.—A draft-gage reading of 2-in. water column means that the pressure of the draft tending to force the outside air into the stack or

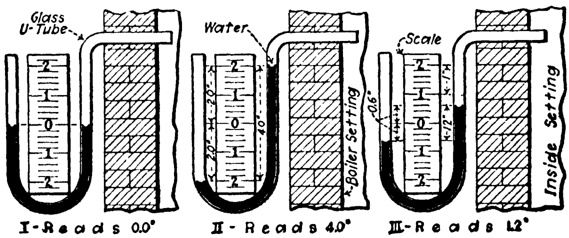


FIG. 360.—Examples in reading manometer draft gages.

boiler, at the point at which the reading is taken, is just sufficient to support a column of water 2 in. high.

504. An elementary manometer or draft gage (Fig. 360) is merely a glass tube of any convenient diameter and having a "U" bent in one end. The U-portion is filled partially with water. One end of the tube is extended into the enclosed

space in which it is desired to measure the draft. The other end is open to the atmosphere. When there is no difference in pressure between the two ends of the tube, the water will rest at the same level (Fig. 360, I) in both legs. If the atmospheric pressure tends to force air into the enclosed space, the water in the leg which connects into the space is forced up (Fig. 360, II and III) correspondingly. The weight of the cool-atmosphere column (W_A, Figs. 361 and 362) acts against

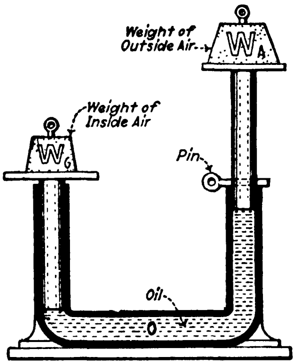


FIG. 361.—The principle of draft. (If the pin is removed, W_A will force W_G up, just as the cold outside air forces up the water in one leg of a manometer.)

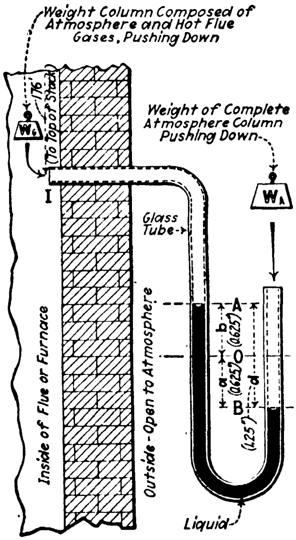


FIG. 362.—The principle of the manometer draft gage.

the weight of the lighter hot-air column and forces it up until the unbalanced height of the water column just equals the difference in the weights.

505. The total static draft, sometimes called the *theoretical maximum static draft,* developed by a chimney is the total pressure difference which results owing to the difference in the weights of the column of hot flue gas inside the chimney and a column of the outside air of the same area and height.

Example.—The total static draft will be indicated by a draft gage, which is located at the grate level, when the ashpit door is closed and no flue gases are flowing. In Fig. 363 and Table XVII (which will be

TABLE XVII.—CHIMNEY DRAFT, DRAFT DROP, AND AVAILABLE DRAFT

I Element of flue gas path	Draft gages: II Identifying letter	Draft gages: III Assumed height of exploring tube above grate, ft.	*IV Elevation draft pressure, in. water column, due to hot gas column	Ashpit door open (all values in in. water column) — Drop in draft due to friction and velocity: V Between successive tube locations	Drop in draft due to friction and velocity: VI Total up to designated location	Available draft: VII Between successive tube locations	Available draft: VIII Total up to designated location VIII = IV − VI	Identifying letter
Outside	A	0.0	1.250		1.250		0.000	A
Boiler								
Ashpit	B	0.0	1.250	*a* to *b* −0.010	1.240	*a* to *b* +0.010	0.010	B
Furnace	C	5.6	1.215	*b* to *c* −0.440	0.800	*b* to *c* +0.405	0.415	C
End 1st pass	D	16.0	1.150	*c* to *d* −0.130	0.670	*c* to *d* +0.065	0.480	D
End 2d pass	E	0.0	1.250	*d* to *e* −0.130	0.540	*d* to *e* +0.230	0.710	E
End 3d pass	F	16.0	1.150	*e* to *f* −0.130	0.410	*e* to *f* +0.030	0.740	F
Inside damper	G	16.0	1.150	*f* to *g* −0.010	0.400	*f* to *g* +0.010	0.750	G
	H	16.0	1.150	*g* to *h* −0.015	0.385	*g* to *h* +0.015	0.765	H
	I	16.0	1.150	*h* to *i* −0.014	0.371	*h* to *i* +0.014	0.779	I
	J	16.0	1.150	*i* to *j* −0.014	0.357	*i* to *j* +0.014	0.793	J
Breeching	K	23.6	1.102	*j* to *k* −0.065	0.292	*j* to *k* +0.017	0.810	K
	L	29.2	1.067	*k* to *l* −0.065	0.227	*k* to *l* +0.030	0.840	L
	M	29.2	1.067	*l* to *m* −0.014	0.213	*l* to *m* +0.014	0.854	M
Chimney								
	N	29.2	1.067	*m* to *n* −0.000	0.213	*m* to *n* +0.000	0.854	N
	O	50.0	0.937	*n* to *o* −0.030	0.183	*n* to *o* +0.100	0.754	O
	P	75.0	0.781	*o* to *p* −0.031	0.152	*o* to *p* −0.125	0.629	P
	Q	100.0	0.625	*p* to *q* −0.030	0.122	*p* to *q* −0.126	0.503	Q
Active	R	125.0	0.469	*q* to *r* −0.031	0.091	*q* to *r* −0.125	0.378	R
	S	150.0	0.312	*r* to *s* −0.030	0.061	*r* to *s* −0.127	0.251	S
	T	175.0	0.156	*s* to *t* −0.031	0.030	*s* to *t* −0.125	0.126	T
	U	200.0	0.000	*t* to *u* −0.030	0.000	*t* to *u* −0.126	0.000	U
	V	0.0	1.250	*u* to *v* +0.213	0.213	*u* to *v* +1.037	1.037	V
	W	16.0	1.150	*v* to *w* −0.000	0.213	*v* to *w* −0.100	0.937	W
Inactive†	X	25.0	1.094	*w* to *x* −0.000	0.213	*w* to *x* −0.056	0.881	X
	N	29.2	1.067	*x* to *m* −0.000	0.213	*x* to *n* −0.027	0.854	N

* Since the total static draft developed by this 200-ft. high chimney is 1.25 in., the drop due to decreasing height of flue-gas column, per foot height, is: 1.25 ÷ 200 = 0.006 25-in. water column.

† It is here assumed that the *inactive* portion of the chimney contains flue gas at the same temperature as that of those in the *active* portion.

discussed later) the total pressure is 1¼ in. (1.25 in.) water column, as indicated by gages B, E, and V. Figure 364 shows conditions with the ashpit door open.

506. Chimney draft, draft drop, and available draft have been computed for an imaginary exaggerated set of conditions (Fig. 364) to illustrate the principles involved, and are shown in Table XVII.

507. The formula for computing the chimney height necessary for development of a given total static draft is stated below. Its derivation is too lengthy for inclusion here. The formula is merely an expression, for the difference in weight of unit columns of *hot chimney gas* and of *cool outside air*, reduced to pressure, in inches water column:

$$P_D' = 0.52L_hP_2\left(\frac{1}{T_o + 460} - \frac{1}{T_G + 460}\right) = \text{(total draft, in. water column)} \quad (58)$$

$$L_h = \frac{P_D'}{0.52P_2\left(\dfrac{1}{T_o + 460} - \dfrac{1}{T_G + 460}\right)} = \text{(height, ft.)} \quad (59)$$

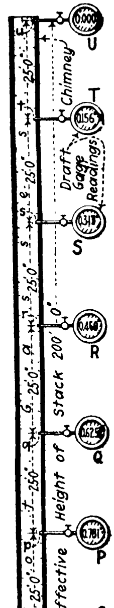
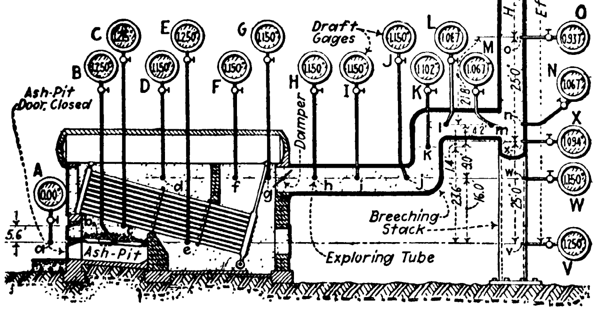



FIG. 363.—Draft-gage readings for boiler setting and stack with fire door closed air tight (this shows a theoretical, ideal condition unattainable in practice).

where P_D' = total static draft, in inches water column, exerted at the base of the chimney or portion thereof of height L_h.

L_h = height of the chimney (or portion thereof) under consideration, in feet.

P_2 = pressure of the atmosphere at the altitude at which the chimney is installed, in pound per square inch = 14.7 lb. per sq. in. at the sea level.

T_o = average temperature of outside air, in degrees Fahrenheit; this is usually taken, in the temperate zone, as about 60 to 65°.

T_G = average temperature of chimney gas, in degrees Fahrenheit.

Example.—What total static draft will a chimney, which is located at sea level and 200 ft. high, produce if the average temperature of the flue gases is 445°F. and an average outside-air temperature of 60°F. is assumed? *Solution.*—Substitute in formula (58): $P_D' = 0.52 \times L_h \times P_2 \{[1/(T_o + 460)] - [1/(T_G + 460)]\} = 0.52 \times 200 \times 14.7\{[1/(60 + 460)] - [1/(445 + 460)]\} = 1.25$-in. water column. Note that the proportions of this chimney correspond with those in the example of Fig. 363 and Table XVII.

NOTE.—The height of a chimney determines the draft which it will develop under given altitude and outside-air and flue-gas temperature conditions. To develop a certain total static draft a chimney should be the

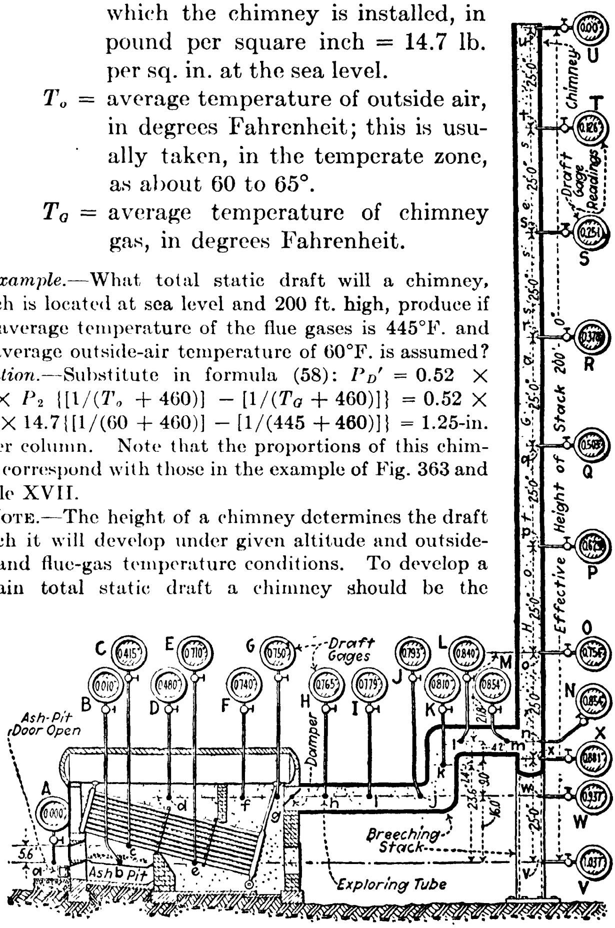


FIG. 364.—Showing draft-gage readings in a specific case with ashpit open.

same height regardless of the number of boilers it serves. But the greater the number of boilers served, the greater the amount of coal burned, the greater the flue area should be. This situation is explained in Sec. 518, wherein is given the formula for computing flue area.

508. There Is a Drop in Draft When Flue Gas Flows in a Chimney.—When the ashpit door is closed and there is no flue-gas flow, there is no drop. Under this condition, assuming that the setting and chimney are airtight, a draft gage (*B*, Fig. 363), which is located at the grate level, will read the *total static draft* (Sec. 505) which the stack develops. In Fig. 363, this is 1.25-in. water column. When flue gas is flowing, the grate-level draft-gage reading (*B*, Fig. 364) will, because of drop, be less than the total static draft. In Fig. 364, it is 0.10 in. Probable values of drop in chimney draft may be predicted, as explained hereinafter, by using data (Table XVIII) which are based on experience.

NOTE.—Drop or loss in draft is similar to the drop in hydraulic pressure which occurs in a pipe when water flows therein. It is similar to the drop in electric pressure (voltage) in an electric circuit which is occasioned by a flowing electric current.

NOTE.—Draft drop is caused by (1) the frictional resistance offered by the rough interior surfaces of the chimney and gas passages to the flue-gas flow (columns V and VI in Table XVII), (2) the imparting of velocity to the flue gas (columns V and VI in Table XVII). In chimney-design practice, cause (2) is usually negligible. Hence in Table XVII, (1) and (2) are combined.

509. The available draft, at any location along the flue-gas path, is the draft which is unexpended, remaining or available at that location. Draft gages indicate available draft. To obtain the available draft at any point along the flue-gas path, subtract from the *total static draft* (Sec. 505) developed by the stack at that location the *total draft drop* up to that point.

Example.—In Table XVII (see also Fig. 364), the total drops are given in column VI. In column VIII the available drafts are recorded. The available draft (*i.e.*, the draft gage reading) at any location = (total static draft at that location) − (total drop to that location). Thus at location *G*, available draft = 1.150 − 0.400 = 0.750 in. water column.

510. Drop in available draft is the drop or decrease in the available pressure difference at successive locations along the flue-gas path. Drops in available draft between different locations may be obtained by taking the difference between the draft-gage readings taken at those locations. Tables,

such as that which follows (Table XVIII) which show approximate drops in pressure along the flue-gas patn, practically always give *drops in available draft.* Such tabulated values (Table XVIII) are based on draft-gage readings observed in operating installations.

Examples.—In Table XVII, which gives values of draft for the conditions of Fig. 364, the drops in available draft, between the different tube locations, are listed in column VII. These between-location drops are merely the differences between the successive available-draft values of column VIII. Note that from *A* to *M* the drops are positive (+), while from *N* to *U* they are negative. This situation will be referred to later. The total available-draft drop from *A* to *M* is 0.854 − 0.000 = in. water column. Also, the total available draft drop from *N* to *U* is 0.854 in. water column.

511. Approximate Drops in Available Draft Which May Be Expected in Boiler Practice.—The values shown in Table XVIII are averages of many determinations. While, in unusual cases, drops differing widely from those tabulated may be encountered, the values given are, ordinarily, sufficiently accurate for estimating the height of a chimney required to satisfy a given set of conditions. Values given are for normal boiler load on the basis of the boiler rating. The drops increase as the load which the boiler is carrying increases.

512. The Drop in Available Draft through Any Element of a Flue-gas Route Represents the Draft Necessary to Force the Flue Gas through That Element.—Hence to determine the effective or net draft which a chimney for a given installation must produce at its base, it is merely necessary to add together the available-draft drops through the furnace, boiler, economizer, air preheater, and smoke conduit. Their sum, in inches water column, will be equal to the net or effective draft which the chimney must develop. How this principle is applied in actual chimney design is explained hereinafter.

513. The "smoke-conduit connection," as the term is used herein, is the location where the smoke conduit from the boiler or boilers joins the chimney. In Fig. 364, the smoke-conduit connection is at *m*.

514. The effective draft of a chimney is the available draft (Sec. 509) developed by the chimney at the smoke-conduit connection, when the flue gas is flowing. It is that

portion of the total static draft, developed by the chimney from the smoke-conduit connection up, which is not lost in the chimney in overcoming friction and in imparting velocity

TABLE XVIII.—DRAFT LOSS IN PRACTICE

No.	Element of flue-gas path	Approximate available-draft drop, in. water column
	Boiler	
1	Ashpit	Often disregarded; should not exceed 0.01 in.
2	Fuel bed and furnace	Depends on kind and size of coal, lb. burnt per hr., design of furnace, etc.; for typical values, see Fig. 365
3	(a) First pass	Depends on: (1) type of boiler, (2) method of baffling, (3) rating; may vary between 0.10 and 8.00 in.
4	(b) Second pass	
5	(c) Third pass	
	Smoke conduit (breeching)	
6	Straight runs	Unlined sheet steel, 0.001 in. per ft. length; brick or masonry, lined, 0.002 in. per ft. length
7	Turns or elbows	Each right-angled turn, unlined steel or masonry—lined, 0.05 in.
8	Damper	Small; can usually be disregarded
9	Economizer and air preheaters	Varies (see Sec. 623) over a considerable range; ordinarily is between 0.10 in. and 2.5 in.
10	Chimney	Is for a chimney of the most economical height and diameter equal to 20 per cent of the total draft (Sec. 505) developed by the chimney, or by the portion of the chimney under consideration

to the flue gas. Therefore, it is the draft that is effective in pulling the flue gas through the boiler and smoke conduit.

Effective draft = (total static draft at the smoke-conduit connection) − (drop in draft in the chimney, from the smoke-conduit connection up) (60)

Example.—In Fig. 364 and Table XVII the effective draft shown at *M* and *N*, is 0.854-in. water column. That is, 1.067 − 0.213 = 0.854 in.

515. Sum of the Available-draft Drops, along the Flue-gas Path from the Ashpit Door to the Smoke-conduit Connection, Must Equal the Effective Draft.—This holds for any operating chimney. Any satisfactory chimney must be of such height that it will develop not only a sufficient draft to circulate the flue gas through itself but in addition it must provide enough excess draft—effective draft—at the smoke-conduit connection to pull the flue gas through the boilers and beechings.

NOTE.—It follows from the above that to determine the *effective draft* which a chimney for a given installation should develop, it is merely necessary to add together all of the draft drops which occur in the boiler, setting, and smoke conduit. The effective draft must equal their sum. Following examples illustrate this principle.

516. To compute the chimney height required for a given installation, the procedure is as follows: First, determine the total static draft which is necessary, by using the following formula, the derivation of which is given below.

$$P''_D = 1.25(A.D.D._{BC}) \qquad \text{(in. water column)} \qquad (61)$$

where P''_D = total static draft which the chimney must develop, from the smoke-conduit connection to its top, in inches water column.

$A.D.D._{BC}$ = available-draft drop (Sec. 510) through the boiler and breeching up to the smoke-conduit connection, in inches water column.

Then to ascertain the height which the chimney should have above the smoke connection to produce the total static draft which is necessary, substitute the value for P''_D determined with formula (61) for P'_D in formula (59) and solve (see following example).

Example.—What height should a chimney be to provide the draft for a boiler installation, when conditions are assumed as follows: Installation is located at sea level. Hence $P_2 = 14.7$. Available draft drops: ashpit, 0.010 in.; furnace and fuel bed, 0.405 in.; first second and, third passes, 0.335 in.; breeching, 0.104 in.; average flue gas temperature = 445°F. Average outside-air temperature = 60°F.

Solution.—First, substitute in formula (61): $P''_D = 1.25(A.D.D._{BC}) = 1.25 \times (0.010 + 0.405 + 0.335 + 0.104) = 1.25 \times 0.854 = 1.067$-in. water column = total static draft which the chimney must develop.

Now by applying formula (59), determine the chimney height necessary to develop a total static draft of 1.067 in water column under the conditions of this example: $L_h = P''_D/0.52P_2[1/(T_o + 460) - 1/(T_G + 460)]$ $= 1.067 \div (0.52 \times 14.7)[1 \div (60 + 460) - 1 \div (445 + 460)] = 174.6$ ft. = height of chimney required above the smoke-conduit connection. Note that this example applies (with negligible error) to the conditions of Table XVII and Fig. 364 for which the $A.D.D._{BC} = 0.854$ in. and the chimney height above the center of the smoke-conduit connection = $200 - 29.2 = 170.8$ ft.

The derivation of the above formula is this: It follows from the statements of preceding Sec. 515 that:

$$E.D. = A.D.D._{BC} \qquad \text{(in. water column)} \qquad (62)$$

where $E.D.$ = effective draft, as defined in Sec. 513, in inches water column. $A.D.D._{BC}$ has same meaning as is specified above. Now, as stated in Table XVIII, the available-draft drop in a chimney may be taken as 20 per cent of the total static draft developed therein. That is, since 20 per cent of the total static draft is consumed in the chimney in overcoming friction and imparting velocity, only $100 - 20 = 80$ per cent of the total draft is available as *effective* draft at the smoke-conduit connection. Hence:

$$E.D. = 0.8\,P''_D \qquad \text{(in water column)} \qquad (63)$$

Now, substituting the value from (63) for its equivalent in (62):

$$0.8\,P''_D = A.D.D._{BC} \qquad \text{(in. water column)} \qquad (64)$$

Simplifying:

$$P''_D = \frac{A.D.D._{BC}}{0.8} = 1.25(A.D.D._{BC}) \qquad \text{(in. water column)} \qquad (65)$$

517. The flue area which a chimney should have should be based on the volume of gas which must be conducted from the boilers. Obviously, the pounds of coal burnt per hour is an important consideration. Furthermore, the flue area of a chimney should be large enough that the gases will be conducted in it without excessive pressure drop (Sec. 510). Otherwise, the effective draft which the chimney develops will be insufficient.

518. A formula for computing chimney-flue area follows. This (as are practically all other chimney-flue-area formulas) is based on practical experience rather than on mathematical analysis. A flue proportioned in accordance with it, will convey the gas with about the 20 per cent drop which is specified in Table XVIII.

$$A = \frac{\mathbf{W}_c}{12\sqrt{L_h}} \quad \text{(area, sq. ft.)} \tag{66}$$

$$d = \sqrt{\frac{W_c}{9.43\sqrt{L_h}}} \quad \text{(diameter, ft.)} \tag{67}$$

where A = cross-sectional flue area of chimney, in square feet.

L_h = height of chimney, above smoke-conduit-connection, in feet as computed by formula (59),

$\mathbf{W}_c$ = coal burnt per hour, in pounds.

d = diameter of chimney flue, in feet.

NOTE.—In using the above formulas for square chimneys, the corners thereof should be neglected. The area taken is that of a circle which would just fit inside the chimney.

Example.—What should be the cross-sectional area of the flue in a chimney which is 140 ft. high and which conducts the gases from a battery of boilers that burn 4,000 lb. of coal per hour. *Solution.*—Substitute in formula (66): $A = \mathbf{W}_c/(12\sqrt{L_h}) = 4{,}000 \div (12 \times \sqrt{140}) = 4{,}000 \div 142.2 = 28.1$ sq. ft. Its diameter would be, substituting in formula (67), $d = \sqrt{\mathbf{W}_c/9.43\sqrt{L_h}} = \sqrt{4{,}000 \div 9.43 \times \sqrt{140}} = 5.98$ ft. $= 5$ ft. $11\frac{3}{4}$

519. The application of the previously discussed principles to the design of an actual boiler-plant chimney will now be explained with an illustrative example, which is taken partly from "*Steam.*"

Example.—Proportion a chimney for a battery of boilers, rated at 2,000 boiler hp., which are equipped with stokers. They burn Maryland semibituminous coal which will evaporate 8 lb. of water, from and at 212°F., per pound of fuel. The ratio of boiler-heating surface to grate surface is 50:1. The unlined steel smoke conduit is 100 ft. long and contains two right-angled bends. The chimney should be capable of handling overloads of 50 per cent. The rated horse power of the boilers is based on 10 sq. ft. of heating surface per horsepower. The chimney will be located at an altitude 2,000 ft. above sea level where the atmospheric pressure is 13.57 lb. per sq. in. The average outside-air temperature may be taken as 60° and the flue-gas temperature, at the 50 per cent overload, as 550°F. The combined grate area of the boilers is 400 sq. ft.

Solution.—The total boiler heating surface $= 10 \times 2{,}000 = 20{,}000$ sq. ft. Hence the grate surface $= 20{,}000 \div 50 = 400$ sq. ft. The total coal burnt per hour (see Sec. 37) $= (2{,}000 \times 34.5) \div 8 = 8624$ lb. Therefore the coal burnt per hour per sq. ft. of grate surface $= 8624 \div 400 = 22$ lb. (approximately). At 50 per cent overload, the combustion rate

will be about 60 per cent greater: $22 \times 1.60 = 35$ lb. per sq. ft. of grate surface per hour.

Now determine the $A.D.D._{BC}$ (refer to Table XVIII):

	Inches Water Column
Drop through furnace and fuel bed with Maryland semi-bituminous coal burned at the rate of 35 lb. per sq. ft. of grate surface per hour as taken from the graph of Fig. 365	0.6
Drop through passes, from Table XVIII	0.4
Drop through 100 ft. of unlined smoke conduit (see Table XVIII) $= 0.001 \times 100 =$	0.1
Drop through two right-angled turns (see Table XVIII) $= 0.05 \times 2 =$	0.1
Total available-draft drop through boiler and smoke conduit.	1.2

Next, determine the total static draft necessary by substituting in formula (61): $P''_D = 1.25\ (A.D.D._{BC}) = 1.25 \times 1.2 = 1.5$-in. water

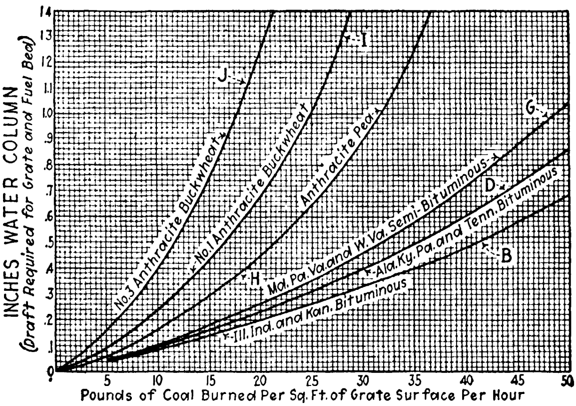


FIG. 365.—Draft pressure required to force air through furnace and full bed for various kinds of coal at different rates of combustion. (*From "Steam."*)

column. Now find the height of chimney required to develop a total static draft of 1.5 in. Substitute in formula (59) $L_h = P'_d/\{0.52P_2[1/(T_o + 460) - 1/(T_G + 460)]\} = 1.5 \div \{0.52 \times 13.57\,[1 \div (60 + 460) - 1 \div (550 + 460)]\} = 1.5 \div \{7.06 \times (0.001{,}925 - 0.000{,}991)\} = 1.5 \div (7.06 \times 0.000{,}934) = 228$ ft. = chimney height above smoke-conduit connection.

The flue area of the chimney should be: $A = W_c/12\sqrt{L_h} = 8624 \div (12 \times \sqrt{228}) = 8624 \div (12 \times 15.1) = 8624 \div 181.3 = 47.5$ sq. ft. The diameter would be: $d = \sqrt{W_c/9.43\sqrt{L_h}} = 7.8$ ft.

520. The draft gages which are used in practice differ from the elementary forms, shown in Figs. 356, 360, and 362, which would not be sufficiently accurate because of the

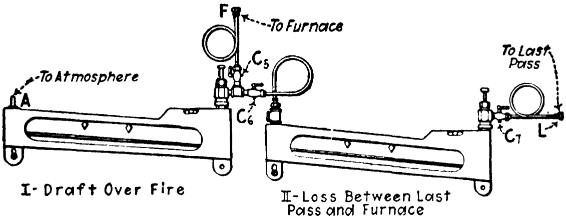


FIG. 366.—Two draft-pressure gages arranged to show: (1) available draft pressure over fire and (2) drop in available draft between last pass and furnace. (If it is desired, in forced-draft installations, to show the draft pressure in the ash-pit, the air vent A may be connected with the closed ash-pit, a three-way cock, similar to those shown, being installed in the connecting pipe.)

short lengths of their scales. In the manometer-type "differential" gages (Fig. 366), ample accuracy is attained by inclining one leg of the gage-glass tube and using in it a light oil instead of water. Thus a long scale is provided. The

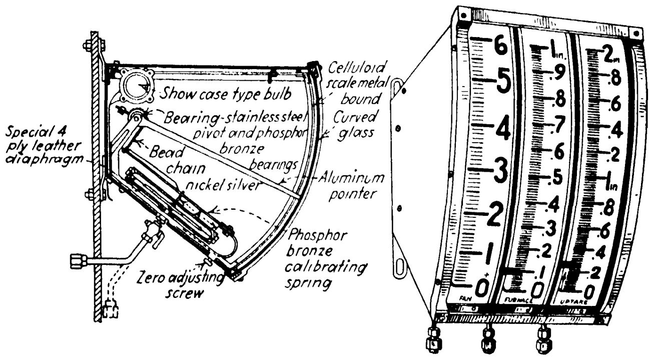


FIG. 367.—Multipointer draft-gage diaphragm operated. (*Hays Corporation.*)

inclined tube permits the liquid to move the same vertical height as when the tube is vertical. But the distance moved through is generally about ten times as great with the inclined as with the vertical tubes. Thus the lengths of the scale divisions are multiplied by 10. This renders the inclined-

tube instruments easier to read. The long inclined scale may be so divided that differences of draft pressure of 0.01-in. water column may be read easily. In the bellows-type gage (Fig. 367) the pointer is actuated, through a lever system, by the movement of the elastic type of a metallic bellows. From the interior of the bellows, a tube connects with the boiler space in which the draft pressure is to be measured. These gages may be either indicating or graphic as shown by the illustrations.

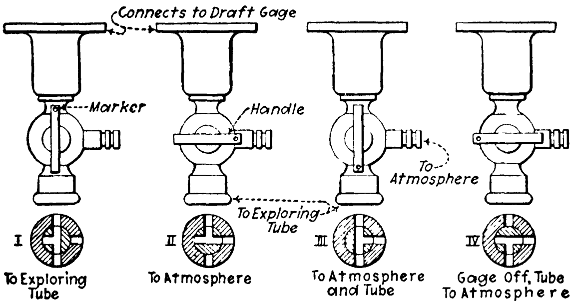


FIG. 368.—Three-way cock used with draft-pressure gages.

NOTE.—Three-way cocks (Fig. 368) may be provided on draft gages (C_5, C_6, and C_7, Fig. 366) whereby the gage may be connected either to the atmosphere or to the exploring tube.

521. In locating draft gages, they should always be so placed that they can be seen by the fireman without effort. When so located the fireman will learn to fire by the draft gage rather than by the steam gage. A good location is on a panel with other controls and instruments.

522. Draft-gage piping may be ⅛- or ¼-in. iron, or copper except that close to the gage it is better to use brass pipe. If iron pipe is used near the gage, rust and scale may drop into the glass. It is often recommended that rubber tubing should not be used for permanent installations. The rubber soon dries out and cracks under the action of the heat.

523. In installing draft-gage exploring tubes (Fig. 369), a length of 1-in. iron pipe T is cemented into the setting.

The tee on the outside end provides connection for the pipe to the gages. If it becomes clogged with soot, the plug can be removed and the obstruction padded out.

524. The location of exploring tubes in furnaces should be as near as is feasible to the front of the furnace and near the top of the chamber so that slag will not accumulate in it.

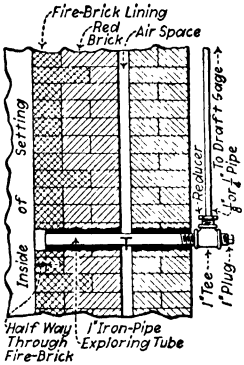


FIG. 369.—Draft-gage exploring tube installed in setting.

525. The best draft to use in the furnace (U. S. Bureau of Mines Bulletin) is that which will satisfy the load conditions and produce the best percentage of CO_2 without CO and without fusing the ash. There is practically no exception to this rule. Determine the proper draft for your plant by using a CO_2 gas analyzer (an Orsat apparatus).

NOTE.—The best draft over the fire for hand-fired furnaces burning bituminous coal ranges from 0.25 to 0.40-in. water column. But it may be materially more or less depending on the grate used, the size and ash content of fuel, and on similar conditions. When forced draft is used overfire draft should be 0.05 in. or as nearly balanced with the atmosphere as possible.

QUESTIONS ON DIVISION 20

1. What is the function of draft in a boiler? Explain.

2. Is a boiler draft due to a pressure or to a vacuum?

3. Write a classification of the different methods of producing draft.

4. Explain the functions of a power-plant chimney.

5. Explain, with an example, the basic principle of chimney draft.

6. Explain, using suitable sketches for illustration, how a power-plant chimney produces a draft.

7. In what unit is draft pressure measured in practice? Explain with an example.

8. What is a manometer? Draw a sketch of one and explain how it works and how to read it.

9. Define total draft. What produces it? Explain with an example.

10. State the formula for computing the total draft which will be developed by a chimney. What do the symbols mean? Same, for formula for chimney height.

11. What is meant by drop in draft? What causes it? Explain in full.

12. Define available draft. How may available draft be determined? Explain with a suitable example.

13. State what drops in available draft may ordinarily be expected through (*a*) ashpit, (*b*) fuel bed and furnace, (*c*) the passes, (*d*) in breeching straight runs, (*e*) breeching elbows, (*f*) damper, (*g*) economizer, and (*h*) chimney.

14. Define effective draft. How is it determined?

15. The sum of the available draft drops along the flue gas path from the ashpit to the smoke-conduit connection equals what? Explain with a sketch and an example.

16. State the formula for computing the chimney height required for a given installation. Explain its derivation.

17. What factors should determine the flue area for a power-plant chimney? What occurs if flue area is insufficient?

18. State the formulas for computing the flue area and diameter of a power-plant chimney. What do the symbols mean?

19. Explain the steps necessary and the formulas used in proportioning a power-plant chimney.

20. What type of draft gages are used in practice? Explain their operating principles.

21. Discuss the locating, installing, and piping of draft gages and exploring tubes.

22. What is the best draft to use over the fire for maximum economy? What are average values of economical draft? What factors affect these values?

PROBLEMS ON DIVISION 20

1. What total draft will be developed by a power-plant chimney which is 110 ft. high and located at the sea level? The average temperature of the outside air is 65°F. and that of the flue gases 550°F.

2. How high must a chimney be to develop a total draft of 2-in. water column, if it is located at an altitude where the atmospheric pressure is 13 lb. per sq. in.? The average outside-air temperature is 55°F. and that of the flue gas 500°F.

3. If, at a certain location in a chimney, the draft due to elevation is 0.47 in. and the friction-and-velocity drop up to that location is 0.09 in., what is the available draft at that location?

4. If the available draft drop up to the smoke-conduit connection, in a certain installation, is 0.75 in., what total draft must the chimney develop from the smoke connection up?

5. What should be the area of the flue of a 120-ft. high chimney which supplies draft for a battery of boilers which burns 1.5 tons (2,000 lb. tons) of coal per hour?

6. Proportion a chimney for the following plant: eight boilers rated at 500 hp. each; grate area of each, 83 sq. ft.; fuel is Illinois bituminous of which 4 lb. is required per rated boiler horsepower hour; proportion stack on basis of a 25 per cent overload; unlined steel breeching is 50 ft. long and contains two right-angled turns. Plant is to be located 2,000 ft. above sea level where atmospheric pressure is 13.57 lb. per sq. in. Temperature of outside air averages 60°F.; that of flue gases 550°F.

DIVISION 21

CHIMNEYS, BREECHINGS, AND DAMPERS

526. A power-plant chimney or stack may be defined as a vertical hollow column, through the interior of which pass the smoke and gas from the boiler furnace. See Sec. 500 for functions.

NOTE.—Strictly, the term *chimney* relates specifically to a masonry structure, while *stack* refers to one of metal. In power plant nomenclature these two terms are often used synonymously.

527. The constructional requirements for chimneys are, in general, two: (1) The proportions of the chimney must be such that it will fulfill satisfactorily its functions for a long period of time; (2) the materials which are used must be such that they may be worked and assembled readily and that they will withstand to a reasonable extent the effects of any elements which may act upon them.

528. A Chimney May Be Built either of Masonry, Steel, or Concrete.—Each of these materials has characteristics which adapt it for certain applications but which may render it unfit for others. Often it is a matter of choice as to the material which should be used. But usually there is a good reason, economic or otherwise, for utilizing a certain material.

529. The Principal Agents Tending to Destroy a Chimney Are the Weather (Including the Wind), Heat, and the Gas of Combustion.—The rain and atmospheric and temperature changes tend to wear away the outside of the chimney. Masonry structures, if properly designed, withstand successfully these destructive elements. Dampness and temperature changes render it difficult to prevent corrosion of a steel stack. If the steel is well protected by a suitable paint, there will be little external corrosion. The interior of the steel stack and the seams may be attacked by certain of the products of combustion. The most important of these is sulphur dioxide.

A portion of it is converted to sulphuric acid, which "eats away" the steel.

530. A masonry stack being thick is relatively cool on the outside, while hot on the inside due to the hot gas. This has a tendency to expand the interior of the chimney more than the outside, thereby pulling apart the exterior of the wall. This may result ultimately in considerable weakening of the structure. In concrete chimneys it is found that the gases, which always carry moisture, are, in passing up the stack, chilled by the relatively cool wall. The moisture is then condensed on the interior of the wall. It enters the fissures and may attack the reinforcing bars. Eventually the stack might fail. Some of the constituents of the gas are thought to attack concrete and impair its strength.

531. All of these interior destructive tendencies, regardless of the material of the stack, are practically eliminated by a suitable lining which may be of concrete, common brick, or firebrick. When the stack is thus lined, it has to withstand only the wind pressure and the destructive tendencies of the natural elements.

532. The Elevation to Which a Chimney Lining Should Extend May Vary from About One-fifth of the Height to the Total Height of the Structure.—The determining factors are (1) the temperature of the gas entering the chimney; (2) the material of which it is built. If the gas is relatively cool, say 300 to 500°F., the lining need not extend to any great height, or there may be none. But if the temperature is very high, say double that specified above, the lining should be extended correspondingly.

533. In designing masonry chimneys, the procedure should be as follows:

1. The height and flue area of the chimney should be determined to satisfy the conditions under which the stack is to operate (Secs. 507, 518).

2. The thickness of the walls should be calculated (Sec. 573).

3. The weight of the chimney should then be calculated from the volume of the material and its weight per cubic foot.

4. The maximum stresses due to dead weight and wind pressure may now be computed for any section. Usually these

stresses must be determined at the planes where the chimney is weakest, such as bed joints, etc. (Sec. 557).

5. The maximum stress on the soil under an assumed foundation is then determined (Sec. 548).

6. If the above conditions are satisfied, then the stability of the structure should be checked as referred to any horizontal section and to the bottom of the foundation (Sec. 550).

534. In checking the design of a self-supporting steel stack the following procedure is recommended:

1. The height and flue area of chimney should be sufficient to insure proper functioning (Secs. 507, 518).

2. The thickness of the steel plate should be ample (Secs. 529, 558) and the riveting properly done (Sec. 567).

3. The maximum stresses due to wind on any weak section should be determined, assuming that failure may be due to buckling (Sec. 558).

4. The maximum stress, due to wind, imposed upon the supporting soil under the foundation should be calculated (Sec. 547).

5. The stability of the structure should be determined with the stack and foundation considered as a unit (Sec. 550).

535. Any chimney should have ample height for creating the draft needed by the boiler and for discharging waste gas at an elevation that will remove all probability of damage to surrounding property and discomfort to occupants of the neighboring premises. Also it should have ample flue area for passing effectively the gas generated from whatever fuel may be burned in the plant. The height and area, when based primarily on draft requirements (exceptions may be encountered in a tall office building, where the chimney height is predetermined), will be determined by methods explained in Secs. 507 to 518.

536. The effect of wind pressure on a chimney is nearly always an important factor. It will now be considered in some detail. Wind blowing against a chimney tends to (1) overturn it as an integral unit, Fig. 370; (2) break it at some plane of least resistance, Fig. 371; (3) throw it out of perpendicular by causing an edge or corner of the foundation to fail, Fig. 372, or to sink it, Fig. 373, into the supporting soil.

To safeguard against failure due to these tendencies, the chimney structure must be properly designed, as will hereinafter be explained.

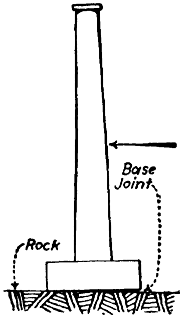


FIG. 370.—Entire structure tilting under wind pressure—unyielding soil beneath foundation.

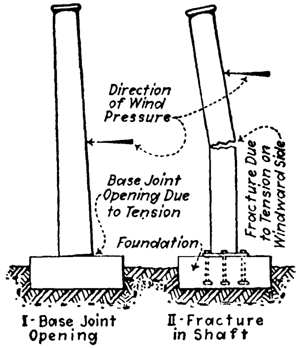


FIG. 371.—Chimney joints opening on windward side due to wind pressure.

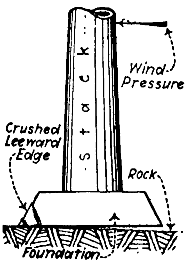


FIG. 372.—A chimney foundation may crush on the edge.

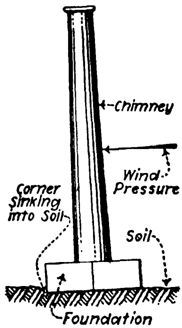


FIG. 373.—Leeward corner of foundation sinking under wind pressure.

537. The maximum wind pressure assumed as acting upon each square foot of projected area of one side of a square chimney is 50 lb. When the chimney is hexagonal, the pressure may be taken as about 40 lb. per sq. ft. of projected area,

when octagonal as between 30 and 35, and when round as between 24 and 30 lb. per sq. ft. In the determination of the above values, it was assumed that the most violent wind never has a velocity exceeding 100 miles per hr. Such velocities have been recorded. Chimneys properly designed on the basis of the above-stated values will, so experience has shown, stand.

538. The computation of the wind pressure acting on a chimney is based on the fact that the pressure due to wind

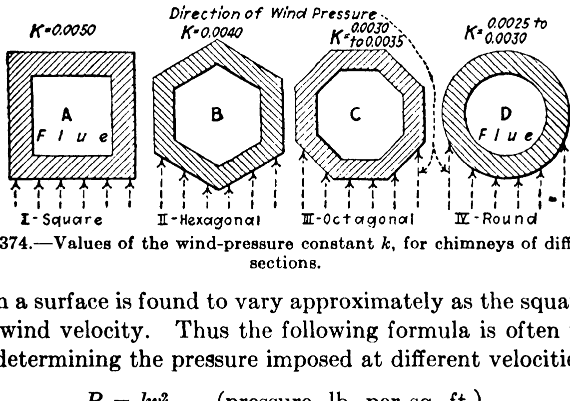


FIG. 374.—Values of the wind-pressure constant k, for chimneys of different sections.

upon a surface is found to vary approximately as the square of the wind velocity. Thus the following formula is often used for determining the pressure imposed at different velocities.

$$P = kv^2 \qquad \text{(pressure, lb. per sq. ft.)} \qquad (68)$$

where P = the pressure due to wind upon a square foot of projected area, in pounds.

k = a constant determined by the contour of the surface against which the wind is blowing.

v = the velocity of the wind, in miles per hour.

On the basis of the unit pressures stated in the preceding section, the values of k which may be safely used (Fig. 374) are: square chimneys, 0.005; hexagonal chimneys, 0.004; octagonal chimneys, 0.003 to 0.0035; round chimneys, 0.0025 to 0.003.

NOTE.—The values of k as computed from data presented by various authorities, vary, for example: Parsons,—square, 0.005; octagonal, 0.00375; round, 0.0025. Peabody and Miller,—square, 0.0055; hexagonal, 0.0041; octagonal, 0.0038; round, 0.003; Rankine—(assuming a value of 0.005 for square chimney) hexagonal, 0.00375, octagonal, 0.003; round, 0.0025. Henry Adams—(assuming a value of 0.005 for square

chimney) octagonal, 0.0041; round, 0.0039. Pratt—square, 0.005; octagonal, 0.0035; round, 0.0025. All the quoted values are high, as it has been shown that, except upon small surfaces, the value of k for a 100-mile-per-hr. wind is only about 0.0032 (Gebhardt) which corresponds to a value of 32 lb. per sq. ft. for a wind velocity of 100 miles per hr.

Example.—A storm wind blowing 75 miles per hr. impinges squarely against a steel stack (Fig. 375-I). What pressure per square foot of pro-

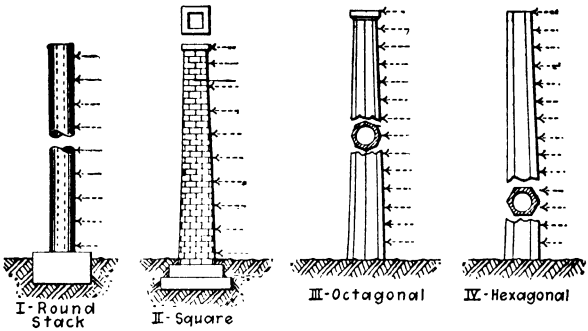


FIG. 375.—Four examples in computing wind pressure against chimneys.

jected area (Fig. 374-IV) does the stack sustain? *Solution.*—By formula: $P = kv^2 = 0.0025 \times 75^2 = 14.06$ lb. per sq. ft. of projected area.

539. Most chimneys, particularly those of masonry, have a taper or batter because the wall thickness may be decreased toward the top without decreasing materially the stability or strength of the structure. The tapered surface is always on the outside. Hence the flue remains of constant diameter from bottom to top. When the stack is tapered, less material is needed. Furthermore, the total force of wind pressure is diminished, since the projected area of the upper portion is smaller than with a straight stack. The taper (Figs. 376 and 377) for the side of a chimney may be approximately $\frac{3}{16}$ or $\frac{1}{4}$ in. to the foot.

540. To compute the total wind pressure against the whole or any section of a tapered chimney, the following formula may be used:

$$F = \frac{L_{wb} + L_{wt}}{2} L_h P \qquad \text{(force, lb.)} \qquad (69)$$

where F = total pressure due to wind, in pounds.
L_{wb} = width of base of chimney, or of section thereof, in feet.
L_{wt} = width of top of chimney, or section thereof, in feet.
L_h = height of chimney, or section thereof, in feet.
P = assumed pressure due to wind, in pounds per square foot, as computed from formula 68.

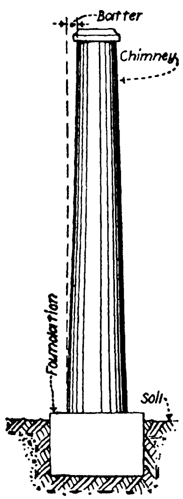


FIG. 376.—Masonry or concrete chimney with uniform batter.

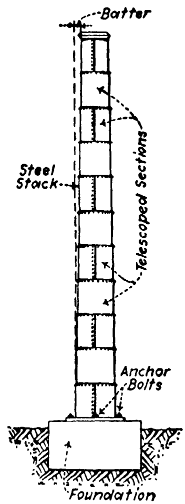


FIG. 377.—Battered steel chimney.

When the chimney is partially shielded by adjacent buildings (Fig. 379), only that portion need be considered which is actually exposed to the wind.

541. The total force of the wind may be assumed to act at the center of gravity of the projected exposed area of a chimney when determining the effect of the wind pressure upon the stability of or upon the stresses in the structure. This point at which the force of the wind may be assumed to act is called the *center of wind pressure.*

NOTE.—The center of gravity of an object is that point within the object at which the weight of the object may be assumed to act. The

center of gravity of a plane surface is the point at which the total force, due to a uniformly distributed force, may be considered as being applied. If a thin sheet of material be suspended at its center of gravity, it will then have no tendency to rotate from any position in which it may be placed.

Example.—From Fig. 378, the total wind pressure, when the wind is exerting a pressure of 14 lb. per sq. ft. $= 50 \times 10 \times 14 = 7{,}000$ lb. which may be considered as being applied as a single concentrated force F at the center of gravity G of the projected area.

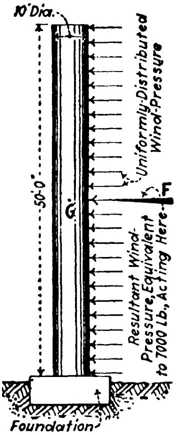


FIG. 378.—Illustrating concentration of wind pressure at center of gravity of projected area.

542. To compute the height of the center of gravity of the projected area of a uniformly tapered chimney, or of any section of it, the following formula may be used:

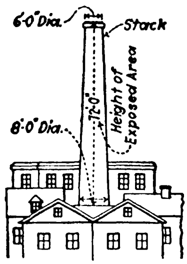


FIG. 379.—Chimney partially shielded from wind by surrounding buildings.

$$L_{hc} = \frac{L_{wb} + 2L_{wt}}{L_{wb} + L_{wt}} \times \frac{L_h}{3} \quad \text{(ft.)} \tag{70}$$

where L_{hc} = height of the center of gravity of the projected area of the chimney or section of it, in feet.

L_h = total height of the chimney or section, in feet.

L_{wb} = width of the bottom of the chimney or section, in feet.

L_{wt} = width of the top of the chimney or section, in feet.

Example.—What is the height of the center of wind pressure above the base of the wind-exposed section of a chimney (Fig. 379) which

extends 72 ft. above the lowest of several buildings which surround it? Its top is 6 ft. across and the base of the section is 8 ft. *Solution.*—Substituting in the above formula: $L_{hc} = [(L_{wb} + 2L_{wt}) \div (L_{wb} + L_{wt})] \times (L_h \div 3) = \{[8 + (2 \times 6)] \div (8 + 6)\} \times (72 \div 3) = 34.3$ ft.

NOTE.—In practice it is often assumed that the center of wind pressure is at one-half the height of the chimney or exposed section. Since the velocity and hence the wind pressure is greater at the top of the chimney than at the bottom, there is very little error in using this method. Such error as there may be is on the safe side.

543. The sustaining ability of the soils on which chimney foundations bear must be considered in the design of any important structure. Practice and experiment have shown that there are certain safe pressures which should not be exceeded. After the weight of a proposed chimney and foundation has been computed, then the weight per square foot which it will impose on the supporting soil should be calculated. If the weight so imposed will be excessive, then the bearing area of the foundation must be made larger so that it will extend over more earth area and thereby decrease the weight imposed per square foot. The following table quotes unit pressures which practice has shown to be safe.

TABLE XIX.—ALLOWABLE FOUNDATION PRESSURES ON SOILS[1]

Kind of material	Safe bearing power			
	Tons per sq. ft.		Lb. per sq. in.	
	Min.	Max.	Min.	Max.
Rock—the hardest—in thick layers in native bed	200	..	2,778	
Rock, equal to the best ashlar masonry	25	30	347	416
Rock, equal to the best brick masonry	15	20	208	278
Rock, equal to poor brick masonry	5	10	69	139
Clay, in thick beds, always dry	6	8	83	111
Clay, in thick beds, moderately dry	4	6	56	83
Clay, soft	1	2	14	28
Gravel and coarse sand, well cemented	8	10	111	139
Sand, dry, compact and well cemented	4	6	56	83
Sand, clean, dry	2	4	28	56
Quicksand, alluvial soils, etc	0.5	1	7	14

[1] (Baker's Treatise on Masonry Construction, p. 342.)

NOTE.—Use the minimum values rather than the maximum. The building laws of a locality may specify the safe values which must be assumed in that locality. If so, such values should be used.

544. On Uncertain Soils the Chimney Foundation Should Be Supported on Piles.—The piles should, preferably, be driven to bed rock. A sandy loose soil usually requires piles. Christie in his "Chimney Design," page 34, states that piles should be of spruce with a diameter not less than 6 in. and driven by a drop hammer weighing one ton or more. The upper ends of the piles are then covered with a concrete footing and the foundation is built thereon. Wooden piles should not be closer together than 2½ to 3 ft. Each wooden pile may be expected to support a load from 8 to 12 tons. Concrete piles may be designed to bear an average of about 30 tons each. (Marks, "Mechanical Engineers' Handbook," p. 1265.) The concrete pile is indestructible and may be driven after molding and seasoning. Or it may be molded in place in a mandrel which has been driven into the ground.

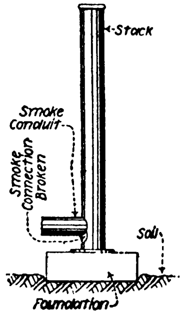


FIG. 380.—Settling of foundation into underlying soil.

545. The Bearing Area of Any Chimney Foundation Must Be Sufficiently Great to Prevent Uniform Settling.—If the pressure per square foot imposed by the dead weight of the stack and its foundation exceeds the values specified in Table XIX, the structure is likely to sink vertically into the supporting soil. This may result in straining and deranging the smoke connection between the boiler and chimney (Fig. 380) or in other damage. When a stack will never be exposed to wind pressure, the bearing area may be determined by applying this doctrine. That is:

$$A = \frac{W_t}{p'_c} \quad \text{(area per sq. ft.)} \tag{71}$$

where A = bearing area of foundation, in square feet.

W_t = weight of the stack and foundation, in tons.

p'_c = pressure imposed by dead weight of stack and foundation, in tons per square foot

p'_c = allowable bearing pressure on soil, in tons per square foot, from Table XIX.

NOTE.—When wind impinges against a stack, the result is to increase the pressure under the leeward side of the stack foundation. Then, the tendency is for the extreme leeward edge to sink (Fig. 373) into the supporting soil. When a stack is to be thus exposed to wind, the foundation area must be so proportioned and distributed that it will safely support the total load due to the dead weight and also that due to wind. Hence a wind-exposed stack must be designed accordingly. How the maximum load, due to the combined dead weight and wind, can be computed will be shown in a following section.

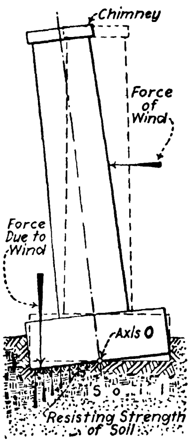


FIG. 381.—Wind pressure is converted to a downward force.

Example.—If a certain chimney and its foundation weighs 85 tons and bear on a soil which will safely support 2 tons per sq. ft., what must be the area of the bearing surface of the foundation? Disregard effects of wind pressure. *Solution.*—Substituting in formula: $A = W_t/p'_c = 85 \div 2 =$ 42.5 sq. ft. which is the area required.

546. When Wind Impinges on a Stack, the Stack Acts Like a Lever Arm and Tends to Force the Leeward Edge of the Foundation into the Soil (Fig. 381).—It is assumed that the chimney in shifting rotates about some point O. (The location of this point need not necessarily be under the center of the foundation as is shown in the illustration.) The bearing strength of the soil under the leeward side tends to prevent this tipping. But if the bearing strength of the soil under the leeward edge is exceeded, the stack will obviously tilt.

547. The formula for computing the maximum pressure, due to wind alone, under the extreme leeward edge of the supporting area of the foundation is as follows:

$$p''_c = \frac{FL_{hc}}{\frac{I}{c}} \qquad \text{(pressure, lb. per sq. in.)} \qquad (72)$$

where p''_c = maximum compressive stress, in pounds per square inch due to wind pressure.

F = force in pounds due to wind considered as applied at the center of wind pressure.

L_{hc} = height of center of wind pressure above the base, in inches.

I/c = section modulus of the supporting area of the foundation. Values of I/c for various areas may be found in handbooks.

Example.—A total wind pressure of 10,000 lb. is considered as applied at 50 ft. above the bottom of a round foundation. What will be the maximum compression due to the wind only, when the base is 16 ft. in diameter? *Solution.*—The value of I/c for a circle is $0.1d^3$ approximately. Then substituting in the formula: $p''_c = FL_{hc} \div I/c = (FL_{hc}) \div (0.1d^3) = 10{,}000 \times (50 \times 12) \div 0.1(16 \times 12)^3 = 8.48$ lb. per sq. in., or 0.61 tons per sq. ft., which is the pressure under the leeward edge. At every other point under the foundation the pressure is less than 0.61 tons per sq. ft.

548. The total maximum unit pressure imposed upon the supporting earth is found by adding that due to dead weight and that due to wind, thus:

$$p_c = p'_c + p''_c \qquad \text{(pressure, lb. per sq. in.)} \qquad (73)$$

where p_c = total maximum pressure, in pounds per square inch.

p'_c = pressure, in pounds per square inch, due to dead weight of chimney and foundation.

p''_c = maximum pressure on extreme leeward edge, in pounds per square inch (formula 72).

549. The Wind May Overturn a Chimney If Its Weight and Supporting Foundation Area Are Not Properly Proportioned (Fig. 370).—In practice the materials used in the foundation and the nature of the supporting soil are such that when failure occurs it will be due either to (1) yielding of the soil under the foundation (Fig. 381), (2) crushing of the extreme leeward edge of the foundation (Fig. 372). Obviously, the chimney will topple over more readily when such failure as this occurs.

NOTE.—When designing a chimney, even if the computations show that the allowable stress upon the chimney material will not be exceeded and that the maximum stress on the supporting soil will be within the

allowable limit, the structure should be checked for stability in accordance with rules given in following sections.

550. The stability of a chimney, or of a portion of it above any plane, i.e., its tendency to remain upright, may be determined by a graphical method by employing the principle of the *composition of forces*. This involves the finding of one force which is equivalent in direction and magnitude to all the forces under consideration.

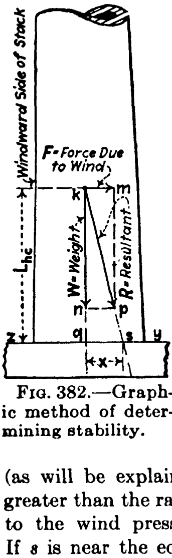


FIG. 382.—Graphic method of determining stability.

Example.—It is desired to determine graphically the stability of a chimney. Refer to Fig. 382, in which the chimney is drawn to some certain scale, say, 1 in. = 1 ft. Then center of wind pressure is located at k. The distance kn is drawn so as to be proportional in length to the weight in pounds of the chimney or of the portion of it above any certain plane, for example zy. The maximum wind pressure in pounds is indicated to scale by the length km. The parallelogram $knpmk$ is then completed and the diagonal kp drawn. The length of kp is then proportional to and gives the direction of the resultant force R. By extending the line of the resultant to the base of the portion of chimney under consideration, the position s on zy is located. If s falls within a certain distance x from the vertical axis q (as will be explained later), the chimney is safe or "stable." If x is greater than the radius of the base, then the chimney has, when subjected to the wind pressure, a tendency to turn over regardless of material. If s is near the edge of the chimney, the leeward side may be crushed and the windward side pulled apart in the tension when the wind blows against the stack.

551. The distance from the axis of the chimney base to the point where the resultant cuts the base may be found from the proportion (Fig. 382):

$$\frac{x}{F} = \frac{L_{hc}}{W} \tag{74}$$

from which the following formula is derived:

$$x = F\frac{L_{hc}}{W} \qquad \text{(distance, ft.)} \tag{75}$$

where x = distance, in feet, from the vertical axis, q, of the section of the chimney to the point, s, where the resultant pierces the section (see the following section for the allowable value of x).

F = force due to the wind pressure, above the plane zy, in pounds.

L_{hc} = distance from the section zy to the center of pressure k of the wind, in feet.

$\mathbf{W}$ = weight of the chimney above the section zy, in pounds.

NOTE.—The above formula follows from the fact that triangle knp is similar to triangle kqs, and $L_{hc} = qk$, $\mathbf{W} = nk$, $x = qs$, $np = km = F$. Therefore, since

$$\frac{qs}{np} = \frac{kq}{kn} \tag{76}$$

then

$$\frac{x}{F} = \frac{L_{hc}}{\mathbf{W}} \tag{77}$$

552. The allowable maximum value for x, or the maximum distance from the axis of the base section of the chimney to the point where the resultant of the wind pressure and the weight of the chimney above the section passes, may be found mathematically, but the following rule is often used: *The resultant should pass not further from the axis of the chimney section than ½ the outer radius plus ¼ the inner radius. For other than round chimneys use the radii of the inscribed circles* (Parsons in "Steam Boilers"). This rule applies to any section of masonry stacks where no tension can be allowed in the outer edge of the wall.

553. For self-supporting chimneys supported on a square foundation, the resultant should pass through the lower surface of the foundation not further from the axis than ⅙ the side of the foundation.

NOTE.—Calculations for concrete stacks, in which tension is always allowable on the windward side, become complicated owing to the combined steel-and-concrete construction. Hence expert reinforced-concrete-stack designers should be consulted concerning concrete stack design.

554. The Thickness of a Chimney Wall Should Be So Proportioned as to Provide a Sufficient Margin of Strength against Crushing, Buckling, and Tension.—This applies to any horizontal cross section from the base upward (Figs. 374 and 375). When wind need not be considered, the stresses set up in the chimney structure are due only to its weight. Hence they are compressive. The essential formulas are given below.

555. The formula for computing the unit stress due to the weight of any chimney above a certain section is:

$$P'_c = \frac{W}{0.7854(d_o^2 - d_i^2)} \quad \text{(stress, lb. per sq. in.)} \quad (78)$$

where P'_c = compressive stress due to the weight of the material above the section in consideration, in pounds per square inch.

W = weight of the structure above the section, in pounds.

d_o = outside diameter of chimney at the section, in inches.

d_i = inside diameter of chimney at the section, in inches.

The expression $0.7854(d_o^2 - d_i^2)$ gives the area of this section.

556. The theoretical formula for computing the unit stress induced at any horizontal chimney section by wind pressure is derived by considering the chimney to act as a cantilever beam. When wind blows against a chimney there is a compressive stress set up in the leeward side and a tensile stress is set up in the windward side. The formula is

$$P''_c = \frac{FL_{hc}}{\frac{I}{c}} \quad \text{(stress, lb. per sq. in.)} \quad (79)$$

where P''_c = maximum stress, either tension or compression, due to wind, at the section under consideration, in pounds per square inch.

F = force due to wind considered as applied at the center of pressure, in pounds.

L_{hc} = height of the center of wind pressure [formula (69)] above the section of the structure, in inches.

I/c = section modulus of the section (the value may be found in handbooks).

The maximum stress occurs only in the extreme leeward or windward fibers or grains of the chimney.

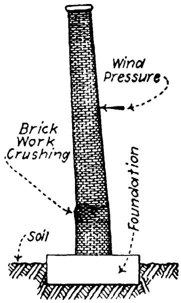


FIG. 383.—Insecure brickwork failing under weight of superstructure and wind pressure.

557. To compute the net compressive or tensile stress in any section due to both weight and wind pressure, the same principle is used as that adopted for computing pressure under the foundation (Sec. 547). It is necessary to know if the compressive stress, due to both wind and weight combined, on the leeward side of the structure is excessive for the material (Fig. 383). This is found by adding P'_c and P''_c. On the windward side the wind has a tendency to produce tension while the weight causes compression. The net result may be either tension or compression. That is, if P'_c is greater than P''_c, then the net stress is compressive and equals $P'_c - P''_c$. If P''_c is greater than P'_c, the net stress is tensile and equals $P''_c - P'_c$.

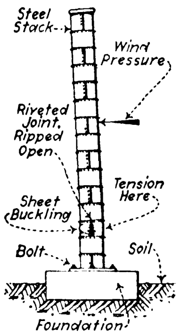


FIG. 384.—Thin ring course failing under weight and wind pressure.

558. When a Chimney Is Built of Thin Material (Such as Steel) and Is Circular in Section, It Will Fail by Flattening or Buckling Rather than by Crushing or Pulling Apart (Fig. 384).—In computing the stress imposed on the plates of steel stacks the weight is neglected. Then the simplified formula used by steel-stack manufacturers is:

$$P'''_c = \frac{FL_{hc}}{0.8d_o^2t} \quad \text{(stress, lb. per sq. in.)} \qquad (80)$$

where P'''_c = stress, in pounds per square inch,

t = thickness of steel plate, in inches. Other

symbols have same meanings as stated above. The safe stress shown in Table XX should not be exceeded.

TABLE XX.—SAFE STRESSES FOR THE VARIOUS MATERIALS USED IN CHIMNEY CONSTRUCTION

Material	Safe stress	
	Lb. per sq. in.	Tons per sq. ft.
Brickwork in Portland cement	208–250	15–18
Brickwork in lime and cement	140–170	10–12
Brickwork in lime	110	8
Radial brick in lime and cement	278	20
Hollow tile	60–80	4.3–5.8
Firebrick { In fire clay	110	8
Firebrick { In air drying or firing cement	170	12
Concrete	208–350	15–25
Steel, single-riveted	8,000	576
Steel, double-riveted	10,000	720
Steel, no rivets	15,000	1,080

559. The weights of material used in building masonry chimneys are about as follows: brick masonry, 120 lb. per cu. ft., concrete, 150 lb. per cu. ft., steel, 489 lb. per cu. ft.

Example.—The brick chimney shown in Fig. 385 weighs about 430,000 lb. What is the unit compressive stress due to weight alone at the bottom, *MN*? *Solution.*—The stress may be determined by substituting in formula 78, thus: $P'_c = W/[0.7854(d_o^2 - d_i^2)] = 430{,}000 \div [0.7854(96^2 - 54^2)] = 87$ lb. per sq. in.

Example.—Find the stress due to wind alone in the outer part of the chimney wall, at *P* and *Q*, Fig. 385, due to a wind pressure of 17 lb. per sq. ft. of projected area when assumed as applied at a height of 45 ft. above the top of the foundation. *Solution.*—Total force, F = projected area × wind pressure = $[(8 + 7) \div 2] \times 96 \times 17 = 12{,}240$ lb. By formula 79: $P''_c = FL_{hc}/(I \div c) = (FL_{hc})/\{0.7854[(r_o^4 - r_i^4) \div r_o]\} = [12{,}240 \times (45 \times 12)] \div \{0.7854[(48^4 - 27^4) \div 48]\} = 84.5$ lb. per sq. in. tension at *P* and compression at *Q*.

Example.—What are the maximum imposed stresses due to weight and wind in the chimney column of Fig. 385 when the wind is blowing? *Solution.*—On the leeward side the total pressure, $P_c = P'_c + P''_c = 87 + 84.5 = 171.5$ lb. per sq. in. On the windward side the stress is the difference between P'_c and P''_c, or $P_c = P'_c - P''_c = 87 - 84.5 = 2.5$ lb.

per sq. in. Since P'_c on the windward side is compression, then net stress, 2.5 lb. per sq. in., is compression. The chimney is safe if the wind pressure is never great enough to change the stress on the windward side to tension.

Example.—Assuming the foundation (*MNRS*, Fig. 385) to be of concrete, 5 ft. deep, and octagonal, 12 ft. across flats, what will be the maximum unit pressure on the earth due to weight and wind? *Solution.*—From preceding example, $\mathbf{W} = 430{,}000$ lb. To find the weight of foundation, multiply the volume by the unit weight. The volume of octagonal prism is V = height $\times 3.314r^2 = 5 \times 3.314 \times 6^2 =$ 596.5 cu. ft. If concrete weighs 150 lb. per cu. ft., the weight of foundation, $\mathbf{W}_1 = 150 \times 596.5 =$ 89,475 lb. The total weight of chimney and foundation $\mathbf{W}_2 = \mathbf{W} + \mathbf{W}_1 = 430{,}000 + 89{,}475 =$ 519,475 lb. The stress per square inch due to weight, $p'_c = \mathbf{W}_2 \div A = 519{,}475 \div (3.314 \times 72^2) = 30.2$ lb. per sq. in. The stress due to wind $= p''_c = (FL_{hc})/(I \div c)$, in which I/c for an octagon $= 0.109d^3$. Therefore $p''_c = [12{,}240 \times (50 \times 12)] \div (0.109 \times 144^3) = 22.6$ lb. per sq. in. The total compressive stress on leeward side, $p_c = p'_c + p''_c = 30.2 + 22.6 = 52.8$ lb. per sq. in. or 3.81 tons per sq. ft., which is safe (Table XIX) for clay. It is to be noted that the stress due to weight is greater than that due to wind, hence there is compression on the windward side, $p_c = p'_c - p''_c = 30.2 - 22.6 = 7.6$ lb. per sq. in.

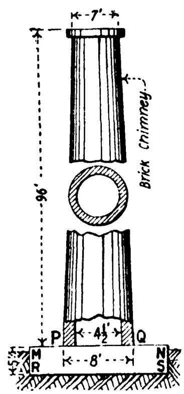


FIG. 385.—Example in chimney design.

Example.—What is the maximum stress on the ground when the foundation (Fig. 385) is circular and 12 ft. in diameter? *Solution.*—The volume, $V = 5 \times (0.7854d^2) = 565.5$ cu. ft. Weight of base, $\mathbf{W}_1 = 150 \times 565.5 =$ 84,825 lb. Total weight of chimney and foundation, $\mathbf{W}_2 = \mathbf{W} + \mathbf{W}_1 = 430{,}000 + 84{,}825 = 514{,}825$ lb. Stress due to weight, $p'_c = \mathbf{W}_2 \div A = 514{,}825 \div (0.7854 \times 144^2) = 31.5$ lb. per sq. in. Since for a circle, $I/c = 0.1d^3$, the stress due to wind, $p''_c = (FL_{hc})/(I \div c) = (FL_{hc})/(0.1d^3) = [12{,}240 \times (50 \times 12)] \div (0.1 \times 144^3) = 24.6$ lb. per sq. in. Maximum stress on ground under leeward edge $= p_c = p'_c + p''_c = 31.5 + 24.6 = 56.1$ lb. per sq. in. or 4.0 tons per sq. ft., which is safe (Table XIX) on good clay.

Example.—If the foundation (Fig. 385) is 12 ft. square, what will be the stress on the ground? *Solution.*—Volume of foundation $V = 12^2 \times 5 = 720$ cu. ft. Weight, $\mathbf{W}_1 = 150 \times 720 = 108{,}000$ lb. Total weight of chimney and foundation $= \mathbf{W}_2 = \mathbf{W} + \mathbf{W}_1 = 430{,}000 + 108{,}000 = 538{,}000$ lb. Stress due to weight, $p'_c = \mathbf{W}_2/A = 538{,}000 \div 144^2 = 25.9$ lb. per sq. in. Stress under leeward edge, when $I/c = 0.118d^3$, $p''_c = (FL_{hc})/(0.118d^3) = [12{,}240 \times (50 \times 12)] \div (0.118 \times$

$144^3) = 20.8$ lb. per sq. in. Maximum stress, $p_c = p'_c + p''_c = 25.9 + 20.8 = 46.7$ lb. per sq. in. or 3.36 tons per sq. ft.

Example.—A steel single-riveted stack is 5 ft. in diameter, 125 ft. high, made of plate $5/16$ in. thick and withstands a wind which exerts a pressure of 40 lb. per sq. ft. against a flat surface. Taking into account only wind pressure, what is the maximum stress set up in the steel? *Solution.*—As wind pressure against a round surface is assumed as $1/2$ that against a flat surface, the effective pressure is 20 lb. per sq. ft. Total force due to wind $= F =$ area $\times$ pressure $= (5 \times 125) \times 20 = 12{,}500$ lb. If the center of wind pressure is $62\frac{1}{2}$ ft. from the base (Sec. 541), $p'''_c = FL_{hc}/0.8d_0^2t = [12{,}500 \times (62.5 \times 12)] \div [0.8(5 \times 12)^2 \times 5/16] = 10{,}416$ lb. per sq. in., which is too high, as only about 8000 lb. per sq. in. (Table XX) is allowed for single-riveted stacks. If the thickness of plate is increased to $7/16$ in. the stress decreases to 7,440 lb. per sq. in.

Example.—If the steel plate in the above example has a uniform thickness to the top, what will be the additional stress due to weight? *Solution.*—It is found that the steel plate necessary to make the stack has a volume of 50.8 cu. ft., which weighs about 25,000 lb. A pressure of about 427 lb. per sq. in. is added. This stress is so small that it is usually neglected.

Example.—If the steel chimney of the preceding example be placed on a foundation 7 ft. deep and 16 ft. square, which rests on sand in its natural bed, will it be safe as a self-supporting structure? *Solution.* The weight, $W = 25{,}000$ lb. Weight of base, $W_1 = 16^2 \times 7 \times 150 = 268{,}800$ lb. The stress on the sand due to weight alone of the chimney and foundation $p'_c = W_2/A = (W + W_1)/A = (25{,}000 + 268{,}800) \div (12 \times 16)^2 = 7.98$ lb. per sq. in. The stress on the sand, due to wind; $p''_c = (FL_{\bar{ho}})/(I \div c) = (FL_{hc}) \div (0.118d^3) = \{12{,}500[(62.5 + 7) \times 12]\} \div [0.118 \times (16 \times 12)^3] = 12.47$ lb. per sq. in. The maximum stress on the leeward side, $p_c = p'_c + p''_c = 7.98 + 12.47 = 20.45$ lb. per sq. in. or about 1.47 tons per sq. ft., which is safe (Table XIX). In the windward side, $p''_c - p'_c = 12.47 - 7.98 = 4.49$ lb. per sq. in. tension, since p''_c is greater. If the stack is lined with firebrick the stress will be increased slightly and the tension on the windward side may be eliminated.

560. The Earth Will Give Away under a Square Foundation When the Wind Is Blowing Diagonally across It Rather than When Blowing Perpendicularly against One Side.—This is illustrated by the following example.

Example.—A certain round chimney has a center of wind pressure 60 ft. above the bottom of its foundation and a force of 14,000 lb. is exerted by the wind. If the base is 15 ft. square, what will be the difference in the maximum stresses when the wind blows diagonally across the foundation and when it blows perpendicular to one side? *Solution.*—When a square rotates about a center line parallel to a side (Fig. 386),

$I/c = d^3/6$. When it rotates about a diagonal (Fig. 386), $I/c = 0.118d^3$. Therefore, for wind blowing perpendicular to one side: $p_c'' = (FL_{hc})/(I \div c) = (FL_{hc})/(d^3 \div 6) = [14{,}000 \times (60 \times 12)] \div [(15 \times 12)^3 \div 6] = 10.3$ lb. per sq. in.

For wind blowing diagonally or against a corner: $p_c'' = (FL_{hc})/(I \div c) = (FL_{hc})/(0.118d^3) = [14{,}000 \times (60 \times 12)] \div [(15 \times 12)^3 \times 0.118] = 14.6$ lb. per sq. in. These values show that the greater (possibly excessive) stress occurs when the wind blows diagonally.

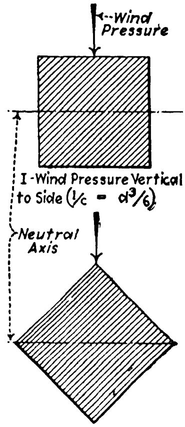


FIG. 386.—Illustrating values for I/c for a square.

561. A steel chimney may often be built inside the boiler house midway of the battery of boilers which it serves. This is feasible, due to the small outside diameter of a steel stack as compared with a masonry stack. Building the stack in the boiler room is an advantage inasmuch as it minimizes the length of the smoke conduit. This insures maximum draft and reduces installation expense. Further space saving may be effected by mounting the stack on the building steel which must then be designed to carry the weight of the stack and the wind thrust.

562. Steel chimneys or stacks may be classified according to the method of supporting them, *i.e.*, (1) guyed stacks, or those which are held upright by guys only; (2) *semi-self-sustaining stacks,* or those which are held upright partially by the foundation and partially by guys (Fig. 387); (3) *self-sustaining stacks* or those wholly supported by the foundation (Fig. 388).

563. A Chimney Which Is Not Self-sustaining Must Be Braced or Guyed.—This is necessary: (1) when the stack is too tall to be self-supporting by anchoring with anchor bolts at the bottom to its foundation; (2) when it is not feasible to provide a base or foundation sufficiently large to support the stack against overturning.

564. If the Stack Is Set upon a Separate Foundation, Usually One Set of Guy Ropes Will Be Sufficient.—Steel ropes are used for guys. Where one set of guys is provided they should be attached at a location about ⅔ the height of

the stack from its base. Where there are two sets, one is attached at about $\frac{2}{5}$ the height and the other at about $\frac{4}{5}$ the height. Where there are three sets they are attached at about $\frac{2}{7}$, $\frac{4}{7}$, and $\frac{6}{7}$ the height. The guy ropes should be attached to guy bands which are riveted to the stack as shown in Fig. 389. Other means of fastening are used but the band is preferable because it affords a strong economical construction.

565. To determine the size guy rope to use for a stack, the following approximate procedure may be followed. Exact rational methods are impossible. Assume that the

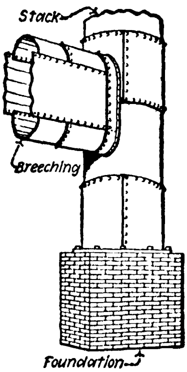


Fig. 387.—Stack with brick foundation.

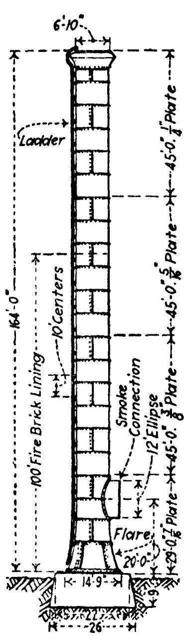


Fig. 388.—Typical example of self-supporting steel chimney.

force due to the wind is supported by only one guy in each set and by the foundation, if it is suitable. That is, if there is one set, assume that one guy and the foundation take all the horizontal force imposed on the stack. If there are two sets, assume that one guy from each set and the foundation resist the force. Compute the horizontal force imposed cn each guy as follows: For the top guy, the load will be the force of the wind pressure (Fig. 390) against all of the projected area above the point of attachment plus that against $\frac{6}{10}$

the projected area between the point of attachment of the top guy and the next lower guy, or the top of the foundation in case there is only one set of guys.

For other than the top guy, where there are more than one set, take $6/10$ the projected area between it and supports or guy-wire attachments above and below. This $6/10$ instead of $1/2$ is assumed to partially correct for irregular initial tension and other indeterminate stresses. Some designers use a value of 0.7 instead of 0.6. Divide the horizontal force, as

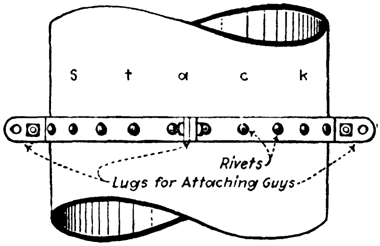


FIG. 389.—Guy band riveted to steel stack.

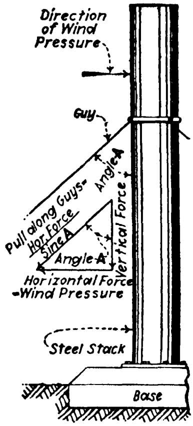


FIG. 390.—Computing tension in guy rope.

above computed, by the sine of the angle (Fig. 390) between the stack and guy. This gives the tension in the guy. To allow for initial tension, add 2,500 to 5,000 lb. for guys from $1/2$ to $7/8$ in. in diameter.

Example.—A steel stack is 150 ft. high and 4 ft. in diameter. It is guyed at 100 ft. from the ground. If the guy slopes 45 deg. from the chimney, what stress must be taken by the guy when the wind pressure is 25 lb. per sq. ft.?

Solution.—Length of projected area is distance above guy attachment $+ 6/10$ distance between guy and foundation or $50 + 60 = 110$ ft. Projected area $= 4 \times 110 = 440$ sq. ft. Horizontal force $= 25 \times 440 = 11{,}000$ lb. Pull along guy (Fig. 390) $=$ horizontal force $\div$ sine A $= 11{,}000 \div \sin 45° = 11{,}000 \div 0.707 = 15{,}500$ lb. or about $7\frac{3}{4}$ tons. From Table XXI a $3/4$ in. rope is required. But since initial tension must be considered as being about 5,000 lb., or $2\frac{1}{2}$ tons, this should be added. $2\frac{1}{2} + 7\frac{3}{4} \times 10\frac{1}{4}$ tons. A $7/8$-in. rope (Table XXI) has a breaking strength equal to the load. In selecting wire guy rope a factor of safety

of 5 to 8 should be applied. In this case the number of guy wires should be increased as the size of wire required is too large for convenient handling.

TABLE XXI.—GALVANIZED IRON RIGGING AND GUY ROPE*
Composed of 6 strands and a hemp center, 7 wires to the strand

Diameter, in in.	Approximate circumference, in in.	Approximate weight per ft.	Breaking strength, in tons of 2,000	Circum. of good grade three-strand manila rope of nearest strength
1¾	5½	4.60	37.00	10
1^{11}/$_{16}$	5¼	4 27	34.70	9½
1⅝	5⅛	3.96	32.40	9
1½	4¾	3 38	27.70	8½
1^{7}/$_{16}$	4½	3.10	25.60	8
1⅜	4⅜	2.84	23.70	7½
1¼	3⅞	2.34	19.90	7
1^{3}/$_{16}$	3¾	2.12	18.10	6½
1⅛	3½	1.90	16.50	6
1^{1}/$_{16}$	3⅜	1.70	14.80	5½
1	3⅛	1.50	13.20	5¼
⅞	2¾	1.15	10.20	4¾
^{13}/$_{16}$	2½	.99	8 86	4¼
¾	2⅜	.84	7 10	3¾
⅝	2	.59	5.30	3¼
^{9}/$_{16}$	1¾	.48	4.32	3
½	1⅝	.38	3.43	2½
^{7}/$_{16}$	1⅜	.29	2.64	2¼
⅜	1⅛	.21	1.95	2
^{5}/$_{16}$	1	.15	1.36	1½
^{9}/$_{32}$	⅞	.125	1.20	1⅜
¼	¾	.090	.99	1¼
^{7}/$_{32}$	^{11}/$_{16}$	.063	.79	1⅛
^{3}/$_{16}$	⅝	.040	.61	1

* From Roebling Catalogue.

566. The Lap of the Riveted Girth Joints of a Steel Stack May Be Upward or Downward.—When the outside lap is downward (Fig. 391), there is a leakage of the soot through the joint to the outside. Acids are carried by the soot and the paint may be destroyed and the stack thereby corroded. A joint of this type (Fig. 391) prevents the rain and snow from

lodging and running to the inside of the stack. It offers minimum resistance to the flow of the flue gases. However, when the courses lap upward on the outside (Fig. 392), the

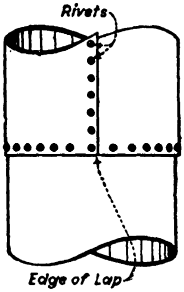


Fig. 391.—Girth joint of stack lapped downward.

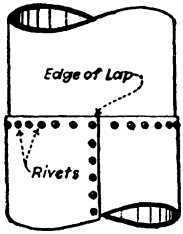


Fig. 392.—Girth joint of stack lapped upward.

joint may be made watertight by using putty of a certain kind. Thereby leakage to the inside is prevented. The joint may be calked if it is required that it be exceptionally tight. To prevent the trickling liquid, which may contain acid and which

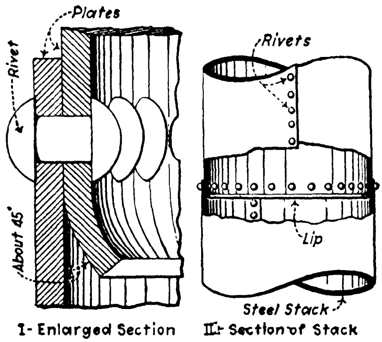


Fig. 393.—Lip turned on edge of inside lap.

runs down the inside of the stack, from being drawn into the joint by capillary attraction, a lip (Fig. 393) is sometimes turned on the uppermost courses.

567. In proportioning the riveted joints in a steel stack, the rules of practice require that the pitch or distance from center

to center of the rivets be approximately 2.5 times the rivet diameter, provided this factor gives a pitch (Fig. 394) less than 16 times the thickness of the plate. If the latter provision is not fulfilled, a factor less than 2.5 must be used. Also, it is required that the rivet diameter be greater than the thickness of the plate, but never less than 0.5 in. Single riveting is usually employed for all joints except the base joint (Fig. 395) where staggered double riveting is used. In stacks of very large diameter all circular seams are double-riveted to insure rigidity.

568. Stone Chimneys Are Seldom Built.—The principles involved are very little different to those relating to brick chimney construction.

569. Brick Are Widely Used for Building Chimneys.—The two types of brick chimneys are: (1) the single-shell (Fig. 396), (2) the double-shell (Fig. 397). The single-shell chimney is serviceable only where brick of an especially good quality and not affected by heat is available. The double-shell, *i.e.*, the lined, chimney is the most common type. The lining is independent of the outer wall thus allowing each wall to expand and contract freely without affecting the other.

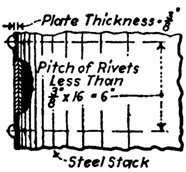


FIG. 394.—Section of vertical seam illustrating maximum allowable pitch of rivets.

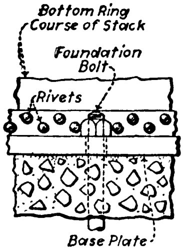


FIG. 395.—Showing double-stagger riveted joint at base of stack.

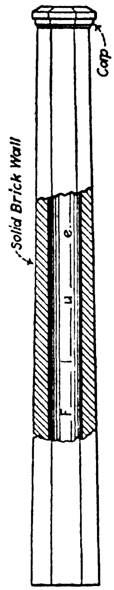


FIG. 396.—Single-shell brick chimney.

570. Buttresses Are Sometimes Sprung from the Outer Shell to Stay the Inner Shell (Fig. 398).—This allows the inner shell to expand without interference. The buttresses

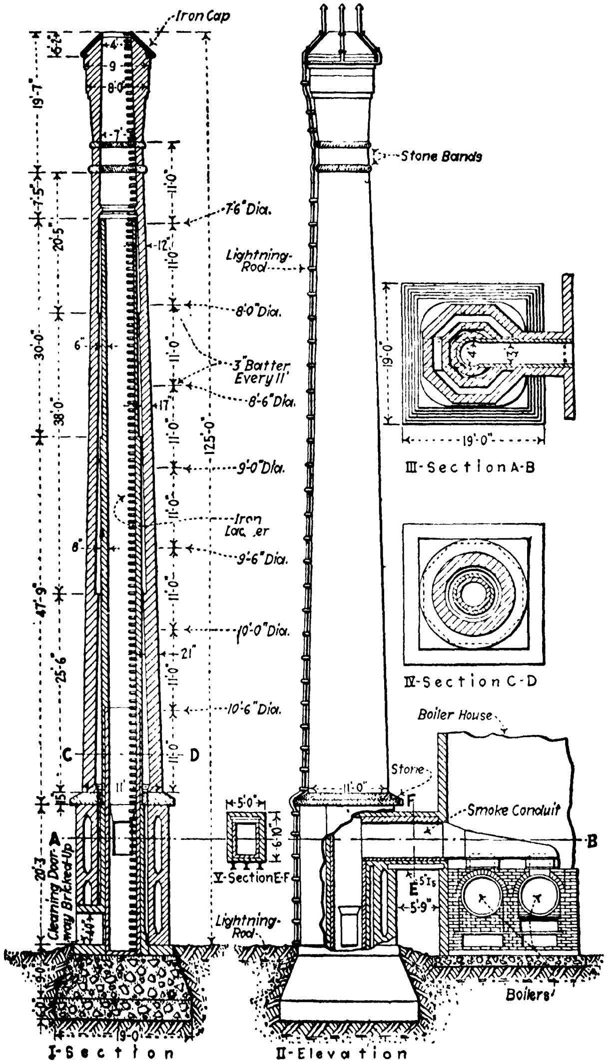


FIG. 397.—Double-shell brick chimney.

should be spaced equally and should stand at least 1 in. clear of the inner shell. There are usually eight of them.

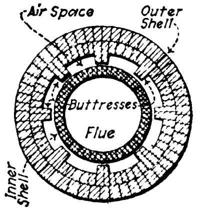


FIG. 398.—Horizontal section of brick chimney.

571. The annular space between the two shells should be at least 2 in. at the top. The batters of the shells should be so adjusted as to increase this distance to 8 or more inches at the bottom.

572. The thickness of brick-chimney walls should be so proportioned as to secure stability with the minimum weight of material consistent with the practical requirements of mason work. Facility of construction requires that the thickness diminish abruptly, from bottom to top, in a series of steps or courses, as indicated in Fig. 399, instead of thinning out gradually.

573. The thickness of the top course of an outer shell built of common brick may be found by the empirical formula:

$$t = 4 + 0.05d_i + 0.0005L_h \quad \text{(thickness, in.)} \quad (81)$$

where t = thickness of top course, in inches.

d_i = inside diameter of chimney at the top in inches.

L_h = height of chimney, in inches. The practical thickness is taken as that nearest to which the chimney can be built with brick.

574. Each succeeding course of 25 or 30 ft., starting at the top of the chimney, should be increased in thickness about 4 in.

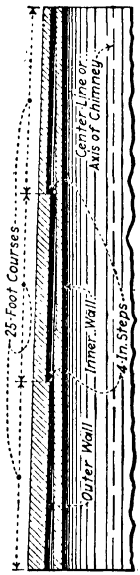


FIG. 399.—showing stepped construction of chimney wall.

Example.—If a chimney is 100 ft. high and the top is 8 in. thick, it might be built of 4 courses each with height of 25 ft. Each course would increase 4 in. in thickness; thus they would be 8, 12, 16, and 20 in. thick.

575. The materials for brick chimneys should be hard-burned close grained bricks for the outer shaft and second-

class firebricks for the core. The outer brickwork from the foundation up to a plane where the winds will have a fair sweep against the chimney should be laid in lime mortar well strengthened with cement. The upper portion should have a certain resiliency to offset the stress of high wind pressure, and this is

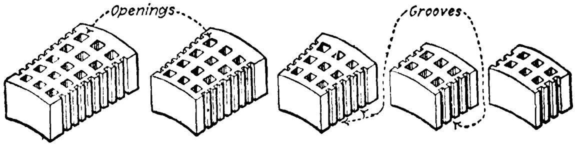


FIG. 400.—Radial brick for chimney building.

best secured by the use of lime mortar containing a smaller admixture of cement. The proportions for the lower part may be 1 part of cement, 3 of lime, and 8 of sand. For the upper part they may be 1–2–6. Since lime does not cling tenaciously to hard, smooth surfaces the harder the brick, the

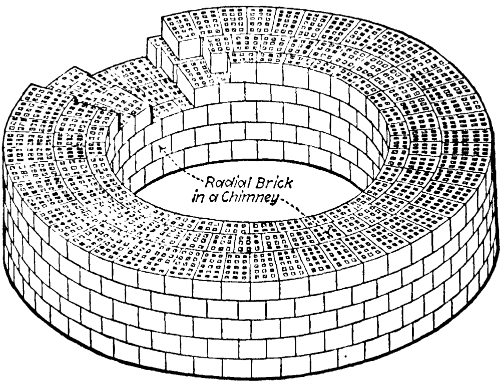


FIG. 401.—Section of chimney.

more the cement that should be used. The core should be laid with pure lime mortar or fire clay. Cement mortar should not be used for any part of the core because it disintegrates in the presence of carbon dioxide and high temperatures.

576. Radial bricks are preferable to common bricks for building round chimneys, as they provide a better appearing and stauncher job. Radial bricks, illustrated in Fig. 400, are made with different radii, to suit all degrees of curvature.

They can be laid (Fig. 401) more compactly than common brick, which require thick mortar joints to fill out the irregularities. A circular column built of these radial bricks makes a very strong structure, because the bricks are virtually keyed in. Each brick in a course acts to hold the entire course intact, like the keystone in an arch. In addition, strength against cracking is added by laying steel bands within the wall.

577. Radial bricks are perforated vertically with square holes. The weight of the brick is thus reduced. Furthermore,

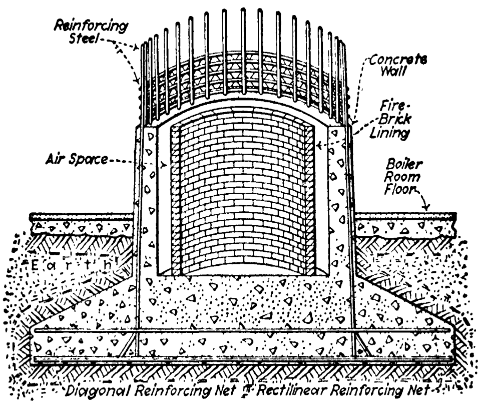


FIG. 402.—Reinforced concrete chimney construction.

the facility which the holes give for thorough burning insures added strength and density. The mortar is worked into the perforations to a depth of about half an inch. The aggregate body of air confined in the multitude of pockets formed by the perforations acts to insulate the structure against heat transference. Hence the radiation is less from a radial brick structure than from one of solid brickwork.

578. Reinforced Concrete Chimneys Are Desirable for Many Reasons.—They may be built rapidly. They (Fig. 402) are strong, because they are reinforced with steel rods which take the tension on the windward side. They are light, thus permitting smaller foundations than are required for brick stacks. They do not occupy so much space as brick chimneys. The interior is smooth, thus minimizing friction.

As concrete chimneys are practically airtight, there is no air leakage which might decrease the effective draft pressure. Disintegration of concrete stacks by the weather is not so noticeable as in steel stacks. The material for construction may be secured in the vicinity, thus reducing transportation expense. Typical construction is shown in Fig. 402. Reinforced concrete chimneys should be specially designed by an experienced concern or engineer.

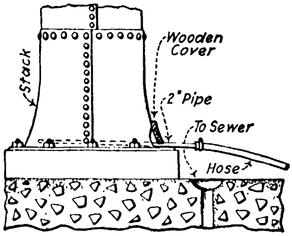


FIG. 403.—Arrangement for flushing out soot from base of stack.

579. Removing the soot from the base of a chimney, where the plant is in continuous operation, should be done with due regard for comfort and cleanliness in the neighborhood of the plant. Flushing the soot out with a stream of water (Fig. 403) is a good way to accomplish this.

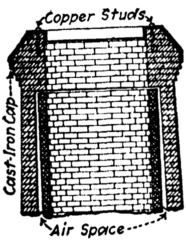


FIG. 404.—Top of brick chimney.

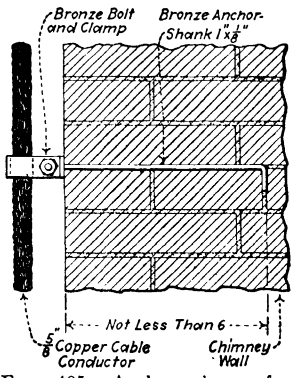


FIG. 405.—Anchor clamp for securing cable to chimney wall.

Example.—A section of 2-in. pipe, attached to a length of fire hose, is inserted through the clean-out opening at the base and extended very nearly to the opposite side. The clean-out door is temporarily replaced with a wooden cover which has an opening at its lower edge large enough for admission of the pipe and for the issuance of the mingled stream of water and soot. As the soot is washed out from the bottom of the pile, the upper surface remains undisturbed.

580. The Top of a Brick Chimney Should Be Protected from the Weather by a Cast-iron Cap (Fig. 404).—At least four

⅞-in. copper studs, for securing the cap, should be left projecting upward from the brickwork at the top. The studs should be riveted over after the cap is set in place.

581. For protecting masonry chimneys from lightning damage (steel chimneys seem to be immune when grounded), the standard specifications adopted by the United States Navy Yard power plants may be followed.

NOTE.—Good specifications require from 2 to 4 lightning conductors,

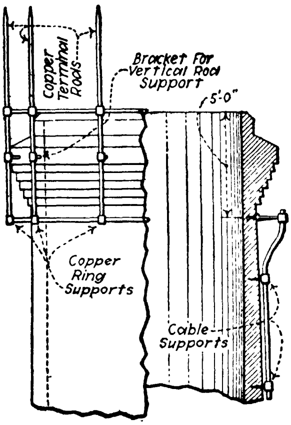


FIG. 406.—Elevation, half section, of chimney top, showing terminal rods.

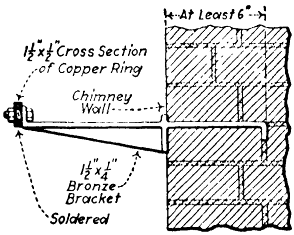


FIG. 407.—Anchor bracket for securing copper terminal ring to chimney wall.

according to the height of the chimney. Where the chimney is 50 ft. or less in height, there are required two conductors; 50 to 100 ft., three conductors; 100 ft. and higher, four conductors. The conductors are of

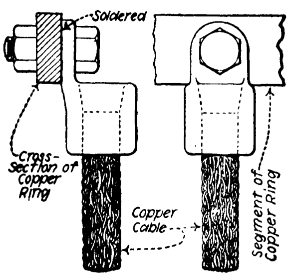


FIG. 408.—Showing attachment of lightning conductor to copper terminal ring.

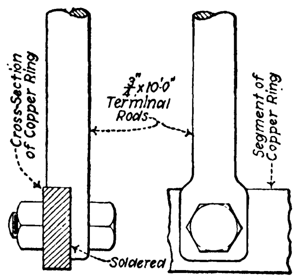


FIG. 409.—Attachment of terminal rods to copper ring.

seven-strand copper cable of approximately ⅝-in. diameter and arranged symmetrically about the chimney. Each conductor is anchored to the

chimney wall by bronze or brass clamps (Fig. 405) in which the cable is clamped at intervals of 10 ft. Every fifth clamp is further secured with solder.

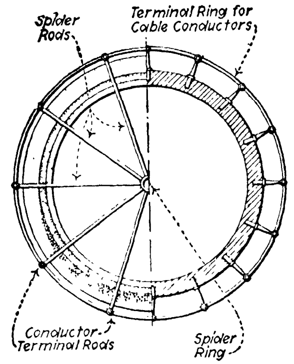


FIG. 410.—Plan of chimney top, half section, showing copper spider for securing terminal rods.

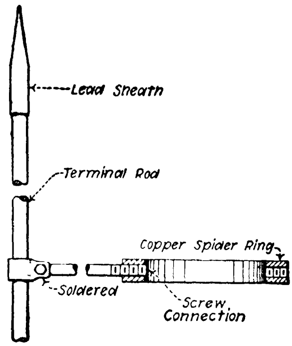


FIG. 411.—Details of copper spider, showing attachment to terminal rod.

At their upper ends, the conductors are attached to a 1½ by ½ in copper ring, which encircles the chimney 5 ft. below the top (Fig. 406) The ring is bolted to bronze or brass brackets which are anchored in the chimney wall (Fig. 407) and spaced not over 2 ft. apart. The cable connections are shown in Fig. 408.

A number of upwardly projecting terminal rods (Fig. 406) ¾ in. in diameter and at least 10 ft. long are attached to the copper ring (Fig. 409) at equidistant locations not over 4 ft apart. These terminal rods are rigidly supported by brackets which are anchored in the masonry (Fig. 406) and supplemented by a copper spider resting on top of the chimney (Fig. 410). The rods of the spider are secured to the terminal rods as shown in Fig. 411.

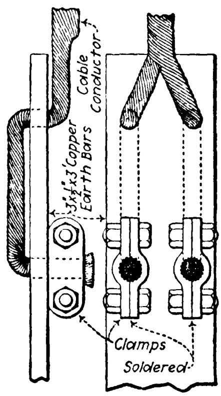


FIG. 412.—Details of earth bar for grounding lightning conductors.

Each terminal rod or *point* is shielded from the corrosive effects of the chimney gases by a thin sheathing of lead which, in accordance with the Navy specifications, extends down about 2 ft. from the point (Fig. 411). At the base of the chimney, each conductor cable is enclosed in a 1½ in. galvanized-iron pipe extending 3 ft. into the soil and 10 ft. above. The cable is electrically connected to the top and bottom ends of the pipe by driving in a metal wedge between the cable and the pipe. The lower end of each cable is securely attached, both mechani-

cally and electrically, to 3 by 1/2 by 36 in. pure copper earth bars (Fig. 412). The earth bars are set below the ground water level—in no case less than 15 ft. below the surface.

582. All Lightning Conductors Near the Top of the Chimney Should Be Sheathed with Lead.—Although some specifications are more liberal, best practice dictates that all copper located above and all within 2 ft. below the top should be sheathed in lead. The specifications of some of the most experienced and prominent engineers so direct.

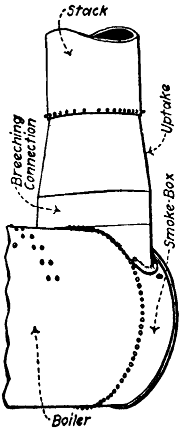


FIG. 413.—Simple breeching.

583. While a square masonry chimney is the easiest to build, it is not desirable because the draft which it handles is not so good as that produced by a chimney of equal flue area and round section. This is due to the greater friction offered by sharp internal corners of the square chimney. The area of a square is not as large as the area of an octagon or circle with the same perimeter. Hence a larger flue is available with same material if the chimney is round in section. A round chimney gives from 1 to 2 per cent stronger draft than a square one of the same flue area.

584. The diameter of the base of chimney is often taken as 1/10 to 1/8 the height. But this should not be accepted as a definite rule.

585. The Term "Smoke Conduit," as Used in This Book, Designates the Smoke Passages between the Boiler or Boiler Setting and the Chimney.—Such connections are designated in various ways by different writers and manufacturers. A smoke conduit may be made of any suitable material. The most common is sheet or plate steel which is formed and riveted into the desired shape. But where the smoke conduit is large or supported on the ground or is underground, it may be of brick, concrete, or other suitable masonry material, which will not be injured readily by the heat and flue gases.

586. The term "breeching" shall be used herein to indicate such smoke conduits as are made of sheet or plate steel.

When the breeching is vertical or at a steep angle as referred to the horizontal, it shall be called an *uptake*. An example

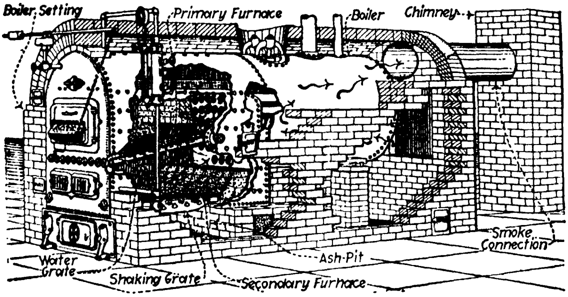


FIG. 414.—Horizontal smoke connection on Kewanee boiler.

is shown in Fig. 413 wherein the stack is connected to the smoke box by the vertical, oval, conical uptake breeching.

587. A "smoke connection" (Fig. 414) is a simple approximately horizontal breeching which leads directly from the boiler into the chimney. Such breechings are used often in heating-boiler installations.

588. An "underground smoke" conduit is a conduit through which the combustion gas is carried underground to the chimney. Such conduits are seldom installed, except where it is necessary to satisfy some special requirement.

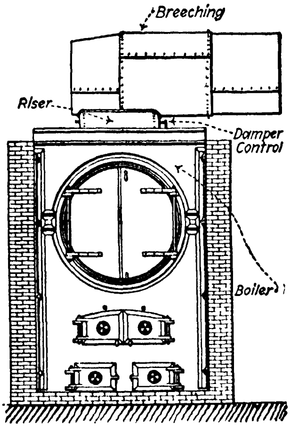


FIG. 415.—Side-breeching for one boiler.

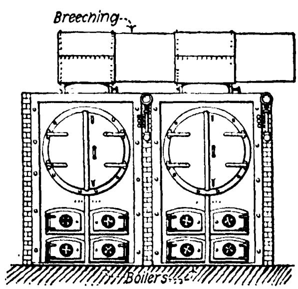


FIG. 416.—Horizontal breeching for two boilers.

589. Underground Smoke Conduits Are Undesirable.—It is difficult to clean them. Furthermore, the gas passing through may be chilled, and hence the effective draft pressure is decreased. The cleaning of the breechings above the ground is easily effected through the doors which may be provided. If water can get into the underground conduit, it will evaporate

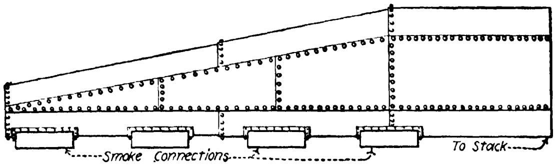


FIG. 417.—Horizontal breeching for four boilers.

and cool the gas as it passes through, thus taking away the draft power of the gas.

590. A common type of breeching connection for one boiler is shown in Fig. 415. It is called a *side breeching.* It leads horizontally to the chimney. The riser may be rectangular or oval in section and the horizontal conduit is circular. Note that the damper is in the riser.

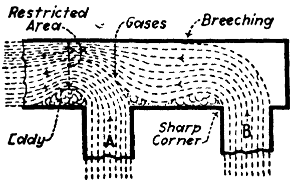


FIG. 418.—Gas path in square-corner breeching. Better draft in *A* than *B*.

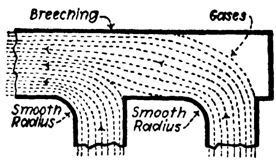


FIG. 419.—Gas path when breeching has curved unions.

591. The arrangement of the breeching where two boilers are served by one chimney (Fig. 416) may follow any one of a number of different designs. Which should be adopted in any given installation must be determined by a consideration of the local conditions which must be satisfied.

592. In Designing Breechings, Outlet Shapes Which Have Sharp Bends Should Be Avoided.—Sharp bends cause an unnecessary loss or drop in draft pressure, making it necessary to install a stack higher than would otherwise be necessary to

overcome their effect (see Div. 20). Figures 418 and 419 indicate the results in a double-outlet breeching when the corners are square and rounded. Figure 420 shows the effect of a sharp angle where an uptake joins a horizontal breeching.

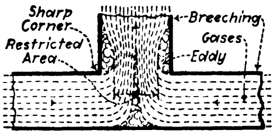


FIG. 420.—Effect of square-cornered riser from horizontal double breeching. Area restricted.

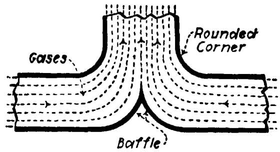


FIG. 421.—Breeching which conducts gases along smooth path.

The area is much restricted and the effective draft pressure is reduced. Figure 421 shows the gas passing without interruption. In large breechings vanes are used at right-angled turns to decrease the friction loss.

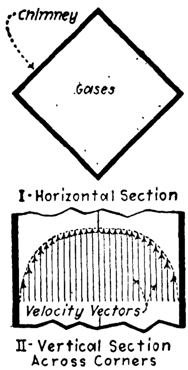


FIG. 422.—Velocity of gases in various parts of square flue.

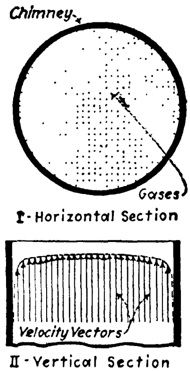


FIG. 423.—Velocity of gases in various parts of round flue.

593. A Smoke Conduit of Round Cross Section Carries More Gas for the Same Area than Does a Square One.—If tested, it would be found that the gas in the corners of the square conduit would be practically stagnant for some distance out from the corner. The curve indicating the velocity of the gas would rise slowly (Fig. 422). This means that the

average velocity of the gases is lowered for the whole area. Consequently, the volume of gas that passes is decreased. In the round conduit (Fig. 423) the gas is stagnant only along a thin film at the periphery of the passage. The velocity curve rises rapidly. The average velocity of the gas is decreased but little. Hence the amount of gas passing, with the same draft pressure, is greater than for an equal square section. Nevertheless the easier construction of rectangular conduit results in its occasional use.

594. The area of the smoke conduit should not be changed abruptly. Such a change will cause a decrease in the draft pressure. The area of the smoke conduit is usually greater than the opening in the chimney by at least 10 per cent. Misostow (*The National Engineer*, Feb. 1913) specifies an area of smoke conduit at least 25 per cent greater than the tube or flue area.

595. Steel smoke conduits should be covered to prevent the radiation of the heat of the gas. When the flue gas is cooled the difference between the density of the gas and that of the outside atmosphere is decreased. Hence the draft pressure is reduced correspondingly. A lining may be placed inside of the breeching but one so located is difficult to repair and keep in place. An outside heat-insulating covering is preferable.

596. Steel smoke conduits are preferable to those of brick or concrete because the friction of the flowing gas on the steel is less than on the other materials. Furthermore, the steel interior surface is smoother and may be cleaned more readily than can brick or the rough masonry.

597. The Thickness of the Steel Plate Used for a Breeching Should Be Determined by Local Conditions.—Ordinarily, breechings are made of No. 8 (0.172 in.) and No. 10 (0.141 in.) U. S. Standard gage steel plate. For small work Nos. 12, 14, and 16 gage plate may be used. Better construction calls for 3/16 in. plate. Some concerns will use no plate thinner than 3/16 in. It costs but little more than lighter stock and has a much longer life, particularly when exposed to extreme conditions. Rivet diameters for plates of the different thicknesses are usually: Nos. 14 and 16—1/4 in.; Nos. 12 and 10—5/16 in.; No. 8, 3/8 in. The pitch is usually about 3 in. unless

the breeching must support an external load in which case correspondingly smaller pitches should be used. Sides of large rectangular breechings are braced or stiffened with angles to prevent vibration. Care should be taken that all seams are well closed, for leakage decreases the effective draft pressure.

598. When the Stack Rests upon the Breeching, the Weight of the Whole Should Not Be Supported by the Boiler Shell.—If the stack is heavy the smoke-box extension may collapse. A stack and breeching support of some description should be used. Figure 424 illustrates such a support the girder of which bridges the boiler. The weight is assumed by the girders and the columns at the sides.

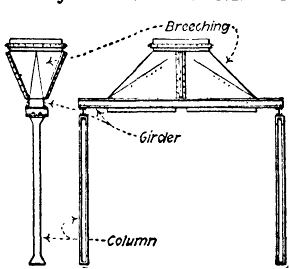


FIG. 424.—Breeching and stack support.

599. Draft Should Be Controlled by Damper Adjustment.—Often combustion is regulated by means of the draft door or by the amount of air that is admitted to the combustion chamber. The resulting combustion may be poor, as compared to that which is possible to attain by damper control.

600. A Damper is usually placed in each smoke conduit or riser, so that the draft in each boiler may be controlled individually. By referring to the accompanying illustrations of breechings, it will be noted that the location of the damper is generally in the outlet from each boiler.

NOTE.—By means of automatic-control devices, which operate through the damper-control lever, the damper may be automatically opened and closed so as to provide the proper draft to insure combustion at a proper rate to maintain the steam-pressure constant or to maintain balanced pressure in the furnace.

NOTE.—The damper is often a casting, or it may be steel plate. Ordinarily ¼-in. plate is used. For the larger sizes, the plate is reinforced with steel angles. Often the damper trunnions turn in bronze bushings.

NOTE.—There is a loss of draft even when the damper is wide open, hence the damper opening should be slightly larger than the flue opening in the chimney.

QUESTIONS ON DIVISION 21

1. Define a power-plant chimney.

2. What is the difference between a chimney and a stack?

3. Are the terms *chimney* and *stack* always understood to mean different things?

4. What two constructional requirements may be mentioned?

5. Of what material may a chimney be built? Why may a certain material be selected?

6. What are the principal agents tending to destroy a chimney? Discuss.

7. How does the heat from the combustion gas affect an unlined masonry chimney with a thick wall?

8. Why is a lining built inside the masonry or steel stack?

9. What determines the height to which the lining should extend?

10. What is the procedure in designing a masonry stack?

11. What calculations and considerations are to be considered in designing a self-supporting steel stack?

12. Why should a chimney be tall and have a large flue area?

13. What are the tendencies due to wind blowing against a chimney?

14. What are the maximum assumed wind pressures that may be employed in designing a chimney?

15. State and discuss the meaning of the formula for determining the wind pressure against a stack.

16. What is "batter" as referred to chimneys? Why have it?

17. How is the total pressure, due to wind, against a chimney calculated?

18. What is meant by the "center of gravity of the projected exposed area"?

19. How is the height of the center of gravity of the projected area determined?

20. If calculations show that the pressure under a foundation of a chimney is too great, what should be done?

21. When the supporting soil is uncertain, what may be done to insure that the foundation will not settle?

22. How is the bearing area of a foundation determined, when the allowable pressure on the soil and the weight of the chimney are known?

23. What is the tendency of a chimney when a strong wind blows against it?

24. State the formula used in computing the maximum pressure, due to wind, under the leeward edge of the foundation.

25. How is the total maximum pressure, due to wind and weight, under the leeward side of foundation determined?

26. If the weight is not sufficient and the foundation not large enough, what would be the result when a strong wind blows against the chimney?

27. How is the distance from the axis of the chimney to the point where the resultant pressure cuts the section determined?

28. For a stable structure how far from the axis of the chimney may the resultant pressure cut the base line?

29. When there is no wind blowing against a chimney, what stress is imposed?

30. How may the compressive stress on each square inch due to dead weight be calculated?

31. If the weight of the stack is neglected, what are the stresses setup when the stack is in a strong wind?

32. How is the total compressive stress on the leeward side of the stack determined? How is the net stress on the windward side determined?

33. If a chimney be constructed of a material that may buckle, what formula is commonly used for determining its safety?

34. When a chimney is resting on a square foundation, will the maximum stress under the foundation be greater when the wind is blowing across diagonally or when blowing perpendicular to one of the sides of the foundation?

35. How may steel stacks be classified?

36. What is the advantage of building a steel stack in the boiler room midway of the building?

37. When is it necessary to brace or guy a chimney?

38. How should the guy ropes be fastened to the stack?

39. Describe the method of determining the proper size of a guy rope.

40. Should the vertical downward force due to the vertical component of the pull exerted by the guy ropes be considered in designing the stack and foundation? Discuss.

41. What may be said concerning the upward and downward lappping of the riveted girth joints?

42. What are the rules for steel stacks concerning the proportions of rivets, rivet spacing, etc.?

43. Are stone chimneys commonly built?

44. May the inner wall of a double-walled brick chimney expand without affecting the outer wall?

45. By what means is the inner wall stayed to prevent it from leaning to one side of the space inside the outer shell?

46. What is the minimum distance to be allowed between the outer wall and the inner wall of a chimney?

47. Is the wall of a brick chimney thinned out gradually toward the top or abruptly? Why?

48. What factors determine the thickness of the chimney wall at the top?

49. What is the usual practice concerning the increase, beginning at the top, in the thickness of a brick chimney?

50. Describe the materials used in constructing a good brick chimney.

51. Describe the radial brick chimney. Is it desirable?

52. Is a concrete chimney a desirable chimney? Why?

53. Describe a good method of removing soot from the base of a chimney.

54. Why place a metal cap on a chimney?

55. Are lightning rods desirable on chimneys? What specifications may be followed in placing rods on the chimney? Describe a typical installation.

56. Why is a square chimney undesirable?

57. What is a common proportion of the diameter of a chimney foundation with respect to the height of the chimney?

58. What is a smoke conduit?

59. Differentiate between the following terms: *breeching, uptake, smoke connection, underground smoke conduit.*

60. Describe a form of "side breeching" for one boiler.

61. Why should a sharp bend in a breeching be avoided? Explain.

62. Why is a round smoke passage better than a square one of equal area? Explain with a sketch.

63. Why is an underground smoke conduit undesirable?

64. As compared to flue area, what should be the area of a breeching?

65. How can the cooling of the gases in a breeching be decreased? Why is this desirable?

66. Discuss the thickness and construction of steel breechings.

67. Why is a damper desirable? Where is it placed?

68. Does the damper, when open, retard gas passing through a breeching?

PROBLEMS ON DIVISION 20

1. An octagonal chimney is in a wind having a velocity of 80 miles per hr. What is the pressure per square foot of projected area?

2. When a pressure of 30 lb. per sq. ft. is applied to the projected area of a chimney which is 120 ft. high, 8 ft. across at the top, and 9½ ft. across at the bottom, what will be the total force against the chimney?

3. Determine the distance from the ground to the center of wind pressure of the chimney of Prob. 2.

4. When the wind exerts a force of 25,000 lb. against a chimney with center of pressure 52 ft. above the bottom of its foundation, what will be the maximum pressure caused by wind alone under a 16 ft. square foundation? (I/c for solid square $= 0.118L^3$ where L = length of one side.)

5. If the chimney and foundation of Prob. 4 weighs 300 tons, what is the total maximum load under the foundation when the wind is blowing as in Prob. 4?

6. Assuming a total wind pressure against a chimney of 30,000 lb. acting at 45 ft. above the foundation and a structure weighing 600,000 lb. what will be the distance from the axis of the chimney to the point where the resultant force cuts the section?

7. A certain chimney weighs 485 tons above a certain section which is circular. The inner diameter is 8½ ft. and the outside diameter 11 ft. What is the stress imposed on each square inch?

8. If the chimney in Prob. 7 is subjected to a wind pressure of 40,000 lb. assumed as at 60 ft. above the section considered, what will be the increased compression on the leeward side? I/c for hollow circular section $= 0.7854\ [(r_o^4 - r_i^4) \div r_o]$.

9. What is the total compressive stress on the leeward side of the chimney of Probs. 7 and 8? Is it safe for brick construction?

10. A certain steel stack is 175 ft. high and 11 ft. in outside diameter. The wind in a certain storm recorded 95 miles per hour. What is the stress set up in the steel and is the stack safe when the material is ⅝ in. thick and single-riveted?

11. The total horizontal force against a guyed chimney due to a strong wind is about 18,000 lb. If this chimney must be held upright by one guy rope which has an angle of 55 deg. with the chimney, what stress must the rope withstand? (Sine 55° = 0.819.) Referring to Table XX what must be the diameter of the rope?

12. Determine by formula the thickness of the uppermost part of a brick chimney which is 225 ft. high and 11 ft. inside diameter at the top. What is the nearest practical value to be used; *i.e.*, a thickness which is a multiple of bricks 2 by 4 by 8 in. laid flat, allowing ½-in. for mortar joint?

DIVISION 22

MECHANICAL DRAFT

601. Mechanical draft to supplement stack draft is required by most boilers installed today because of the rating at which they are operated and also because of the resistance to air and gas flow offered by the combustion equipment and heat-recovery apparatus usually installed. It is common for stokers to require 3 to 4 in. of air pressure under the fire when forced to maximum combustion rates. In industrial plants a draft of 6 to 8 in. is often necessary to overcome the resistance offered by boiler, economizer, air preheater, and cinder catcher and some utility boiler plants have required drafts as high as 13 in. of water. Obviously, it would not be practical to build a stack high enough to produce draft of such magnitude. To meet these pressure requirements, either forced or induced draft is necessary and frequently both.

602. Forced draft supplies air under pressure for combustion. In the case of stokers, the air pressure must be sufficient to force the proper quantity of air through the fire bed to produce the maximum rate of combustion desired. Forced draft is also used with pulverized coal and oil firing when stack draft is insufficient to draw the required amount of combustion air through the burner registers.

603. Induced draft draws the flue gas through the boilers and heat-recovery equipment in the same manner as a stack. It is usually required when either economizer or air preheaters are used because of the resistance they offer to the flow of gas and because the lower gas temperature reduces the draft that the stack can produce. Induced draft is used alone only for gas, oil, or pulverized-coal firing.

604. With forced draft, it is common operating practice to regulate the forced draft to meet the needs of combustion and to control natural or induced draft to maintain a balanced

draft in the furnace. By maintaining a draft of only 0.05 to 0.08 in. in the furnace leakage of air through the boiler setting is reduced and at the same time preventing leakage of furnace gas into the boiler room. Both forced and induced draft are used in all large stoker-fired installations and with pulverized coal, oil, and gas when air preheaters are installed.

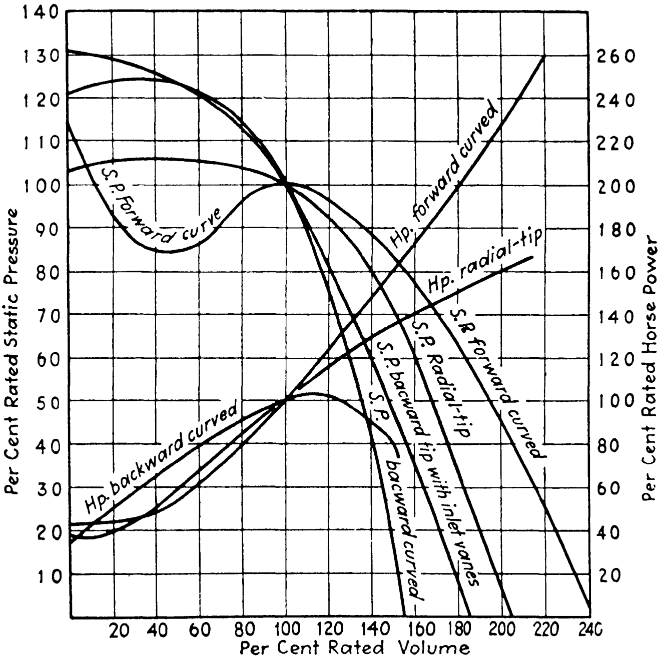


FIG. 425.—Fan-characteristic curves of static pressure and horsepower.

605. Multiblade centrifugal fans are used almost universally to provide forced and induced draft. With forced draft the fan supplies air under pressure. Induced draft fans operate by producing a pressure below atmospheric and so draw the flue gas through the boiler. In small installations a high-speed turbine-driven propeller-type fan is sometimes used for forced draft.

606. Centrifugal fans for induced- and forced-draft service are classified according to (1) the curvature of the fan blades, (2) single or double inlet, (3) direction of discharge. There

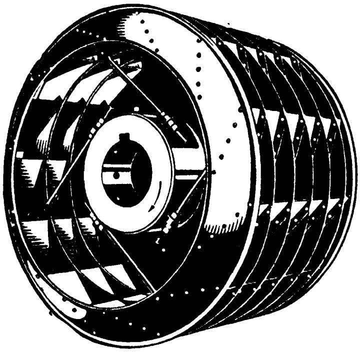


FIG. 426.—High-speed fan wheel for forced draft with backward-curved blades. (*American Blower Company.*)

are four types of fan blading: (1) forward-curved blades, (2) backward-curved blades, (3) Double-curved blades (forward-curve at the inlet and backward-curve at the tip,

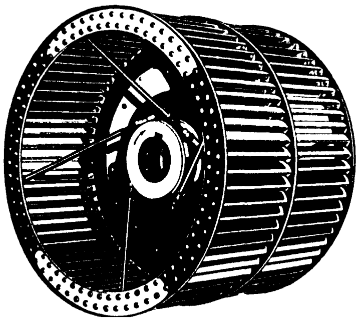


FIG. 427.—Induced-draft fan wheel with forward-curved blades. (*American Blower Company.*)

(4) radial tip. Typical induced- and forced-draft fan wheels and casings are shown in Figs. 426 to 429.

607. Fan Housings.—Within certain size limits, fans for induced and forced draft may be arranged with either single or double inlet. Fans of small capacity usually have single

inlets, and those of large capacity usually have double inlets as in Fig. 428. The fan may be arranged to discharge ver-

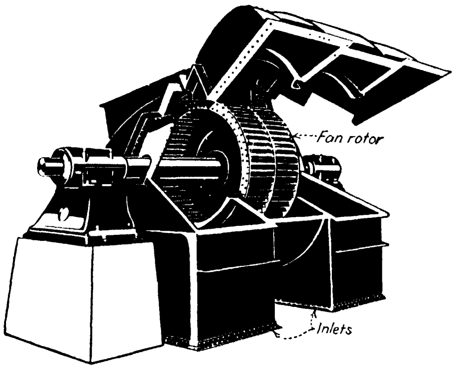


FIG. 428.—Double-inlet induced-draft fan. (*American Blower Company.*)

tically up or down or horizontally or at any of several angles to suit the convenience of local conditions and plant layout.

608. Fan performance is shown by characteristic curves of pressure in inches of water and horsepower plotted against capacity for constant speed operation. Characteristic curves for each of the fan types are shown in Fig. 425. The particular feature of fans with forward-curved blades is a relatively flat pressure-volume curve with a hump in it and a continually rising horsepower curve. The backward-curved fan has a steep pressure-volume curve, and the horsepower curve has a peak or limiting value. Radial tip- and double-curve fans have characteristic curves that lie between these two extremes.

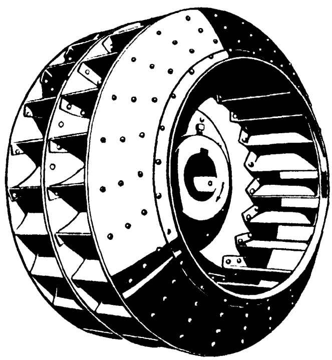


FIG. 429.—Fan wheel with radial tipped blades. (*American Blower Company.*)

609. The horsepower required to drive a fan depends upon the weight of gas handled, the static pressure developed

the density of the gas, and the fan efficiency. The formula for calculating fan horsepower is

$$\text{Horsepower} = \frac{5.19 \times w \times h}{33{,}000 \times d \times E} \tag{82}$$

where w = weight of gas handled per minute.
h = static pressure, in inches of water.
d = density of gas.
E = static efficiency.

Fan efficiency at the design point (peak of the efficiency curve) for the fan types considered ranges between 55 to 74 per cent.

610. For Most Economical Results Fans Should Operate at the Peak of Their Efficiency Curve.—To do this the fan selected must have a capacity-head characteristic that fits the system in which it is to operate and must also be provided with means to control its output to meet the varying demands of boiler operation. The simplest control, but most wasteful of power, is by damper in the fan discharge. The most saving of power is speed control, which is easily accomplished if the fan is driven by engine or turbine. With motor drive, speed control is less simple and with alternating current requires expensive equipment. Speed control may be obtained with constant-speed motors by using variable-speed couplings between fan and motor. Recently adjustable inlet and discharge vanes have been developed that control fan output with relatively little loss in efficiency and are used to considerable extent.

611. Induced-draft fans should always be placed so that the temperature of the gas handled is the lowest possible. This means they should be placed after both economizer and air preheater. This is because, with other conditions equal, the fan horsepower is much higher when handling high-temperature gas than when handling low-temperature gas. Doubling the absolute temperature doubles the fan horsepower. It is usually most convenient to place forced-draft fans on the boiler-room floor or in the boiler-room basement as this results in the shortest connecting duct work. But when air preheaters are installed the forced-draft fan

discharges through the heater and hence is usually placed above the boiler-room operating floor. Sometimes several forced-draft fans are arranged to discharge into a common duct that supplies combustion air to all of the boilers, but more often each boiler has its own fan with a spare arranged so it can serve more than one boiler. Induced-draft fans are rarely connected to a common duct, though frequently two fans operate in parallel to serve one boiler.

612. Main considerations in selecting type of fan for induced- or forced-draft service are the speed and type of the fan drive, static pressure required, and the gas temperature. With these considerations in mind any of the fan types given in Sec. 606 may be the most suitable for either forced or induced draft in a given installation. The main thing is to select a fan that will match the economical or characteristic speed of the drive. For example, if the gas volume is 80,000 to about 250,000 cu. ft. per min. at about 400° and pressure is above 6 in., the forward-curved and radial-tipped blade fans are adapted to direct motor drive because their speed characteristic matches that of the motor. However, above about 12 in. static-pressure wheel-tip speed for the latter fan becomes excessive.

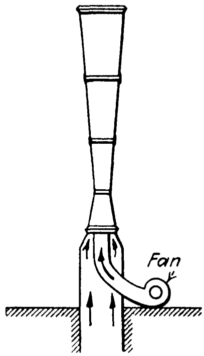


FIG. 430.—Thermix venturi-shaped stack and fan for producing induced draft. (*Pratt Daniels Corporation.*)

But an induced draft fan to handle about 100,000 cu. ft. per min. at a pressure of 1 in. and 500° would require too low r.p.m. for motor drive if a forward-curve or radial-tipped fan were used. Low pressure indicates the use of backward-curved or double-curved blade fans which about double the r.p.m. over the forward-curved blade type and so provide a more advantageous direct-connected motor speed. Because of variations in the fuel bed forced-draft fans operate against fluctuating system resistance. To minimize the variation in air volume delivered through the fuel bed because of this fluctuating resistance, many engineers specify a fan with a steep-pressure characteristic, such as offered by fans

with backward-curved blades. However, forward-curved blade fans also serve as forced-draft fans even though they have flatter pressure characteristic. But they should be selected so that they will operate somewhat to the right of the peak in the characteristic curve.

613. A relatively small fan handling only a portion of the flue gas may be used for induced draft in conjunction with a venturi-shaped stack arranged as in Fig. 430. The fan discharges a jet of high-velocity gas into the throat of the stack causing an injector action.

614. Steam jets are used occasionally to augment draft in small installations and especially in vertical portable boilers that necessarily have short stacks. Also they are sometimes used with chain grates to provide air over the fire.

QUESTIONS ON DIVISION 22

1. Why is mechanical draft equipment required?

2. Define forced draft.

3. What is induced draft?

4. What draft is maintained in the furnace when forced draft is used?

5. What type of equipment is used to produce forced and induced draft?

6. How may fans be classified?

7. Name four types of fan blading.

8. How is fan performance shown?

9. What are the characteristics of fans with forward-curved blades? With backward-curved blades?

10. What is the formula for calculating fan horsepower?

11. How is fan output controlled?

12. Where should induced-draft fans be located?

13. What are the main considerations when selecting type of forced- or induced-draft fan?

14. How may a fan produce induced draft without handling all of the flue gas?

15. When are steam jets used to produce draft?

DIVISION 23

ECONOMIZERS AND AIR PREHEATERS

615. The function of an economizer or air preheater is to absorb heat from the flue gas discharged from a boiler setting. The heat absorbed by an economizer is used to heat feed water, that absorbed by an air preheater heats air used to support combustion.

The heat thus absorbed would otherwise be lost by passing out of the stack into the atmosphere. The economizer and air preheater are more effective in absorbing this *waste heat* than would be additional boiler-heating surface. This is because of the greater temperature difference which affects the heat transfer. Economizers and air preheaters are treated in more detail in the author's "Steam-Power-Plant Auxiliaries and Accessories."

Example.—With 400 lb. per sq. in. gage pressure the temperature of the steam is about 448°F. Hence, with a chimney-flue temperature of 500°, there is only $500 - 448 = 52°$ temperature difference to effect heat transfer from the discharging flue gas to the additional boiler-heating surface. On the other hand, the temperature of water entering an economizer may be 250°. Thus, with an economizer, the temperature gradient which effects heat transfer is at least $500 - 250 = 250°F$.

616. Economizers consist of either curved- or straight-steel tubes through which the feed water passes on its way to the boiler. The tubes are placed in the path of the flue gas usually so that the hottest gas contacts that part of the heating surface carrying the hottest water. Water usually enters at the bottom of the economizer and flows back and forth through tubes connected at each end to return bends or junction boxes. At the top the water from the tube sections is collected in a header and fed to the boiler. Tubes may be plain or have cast-iron fins to increase the heating surface.

617. Integral Economizers Are Placed inside the Boiler Setting.—Some types consist of an upper and lower drum with connecting bent tubes rolled into the drums (Fig. 431). A more recent type is illustrated in the sectional elevation (Fig. 432). It consists of an inlet header into which a large number of tubes are rolled. Each tube forms a continuous

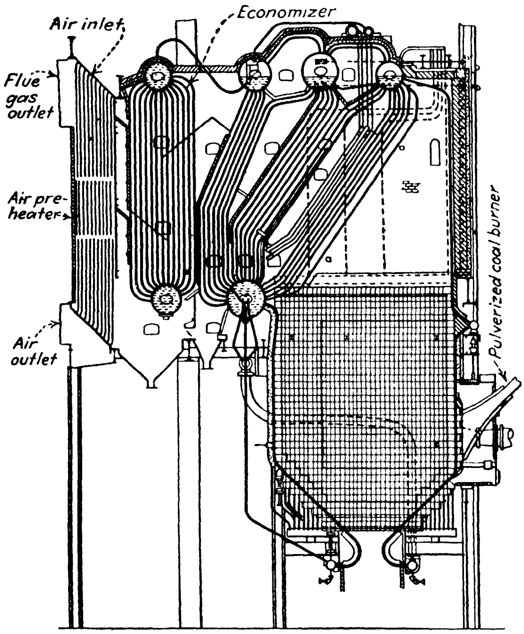


FIG. 431.—Integral economizer and tubular air preheater with bent tube boiler fired with pulverized coal. (*Babcock and Wilcox Company.*)

multi-loop coil and is rolled into the boiler drum at its other end. The multiloop coils are made by welding together a number of bent tubes.

618. Unit or individual economizers (Figs. 433 and 434) have been the type most used though recent high-pressure utility boilers have been equipped with the integral type. Unit economizers must be provided with their own separate casing which is usually of steel, and soot blowers are necessary to keep the heating surface clean.

619. The placing of an economizer with respect to the boiler should be determined by (1) the type of boiler, (2) whether headroom or floor space is more available, (3) whether the

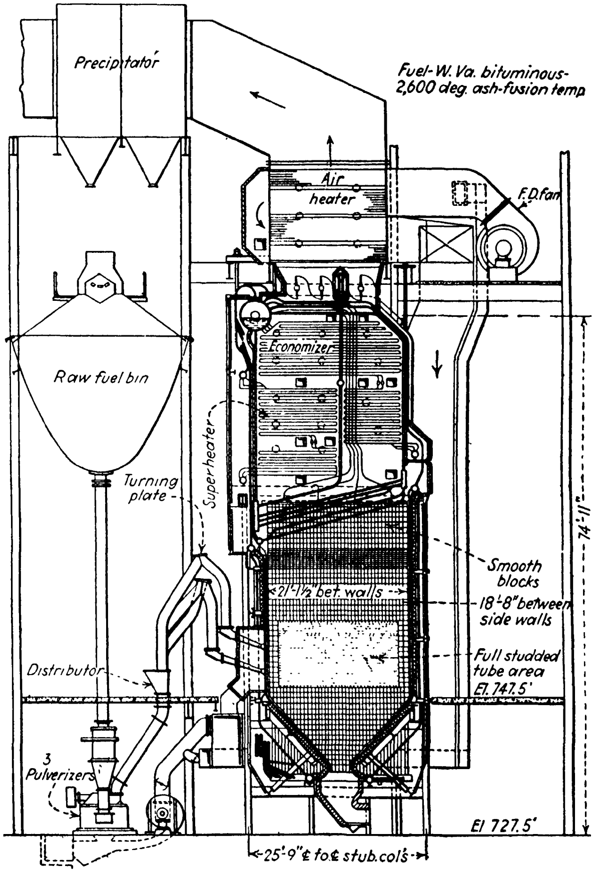


FIG. 432.—Continuous-tube economizer integral with boiler. Gas flow on each side of the vertical baffle is controlled by dampers and gives control of superheat.

installation is in a new or in an old plant. With boilers of certain types, the economizer may be located (Fig. 431) at practically the same general elevation as that of the boiler.

With boilers of other types, the economizer is located most effectively above the boiler. Where the economizer is installed overhead, considerable otherwise-unavailable floor space may be rendered useful. If an air preheater is used in addition to an economizer, the economizer is placed next to the boiler, and the air preheater placed between the economizer and the stack.

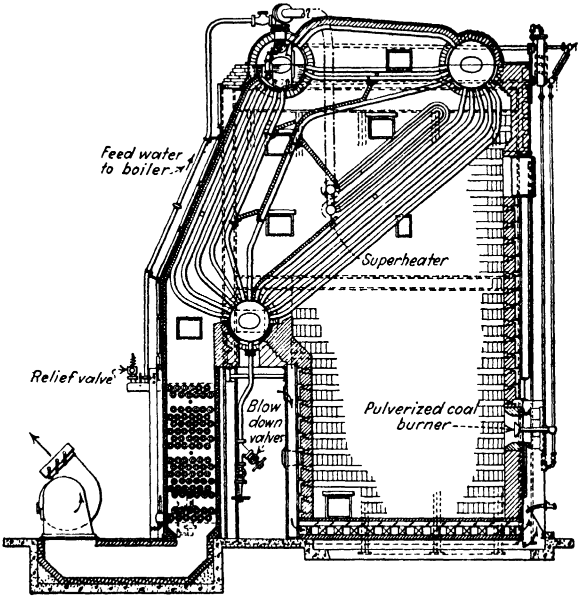


FIG. 433.—Fin tube economizer placed on boiler-room floor.

620. Air preheaters are available in three types: (1) tubular, (2) plate, (3) regenerative. Tubular heaters consist of a large number of tubes rolled into tube sheets at each end and usually placed so the flue gas passes through them. Air being heated is made to make a number of passes across and around the outside of the tubes. Sometimes the gas flows outside the tubes and the air inside. In plate-type preheaters alternate gas and air passages are formed between closely spaced parallel vertical plates. This type of preheater is usually arranged so the gas passes vertically upward through

the narrow passages and air enters at the side and passes downward in one or two passes. The regenerative air preheater (Ljungstrom) consist of concentric rings of alternate flat and corrugated plate arranged so as to form vertical passages for the upward flow of flue gas through one half and the downward flow of air through the opposite half. The elements are slowly rotated so that they are alternately

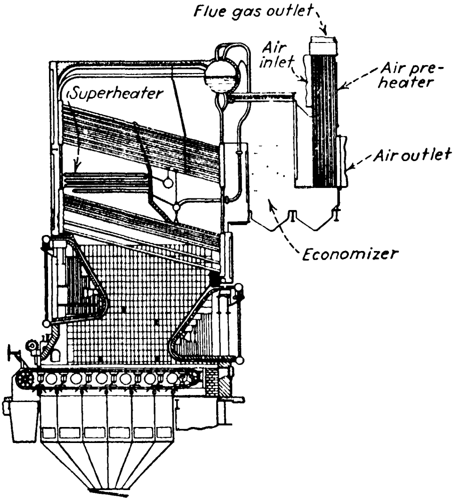


FIG. 434.—Unit economizer with smooth steel tubes and return bends and tubular type air preheater with straight tube boiler fired by chain grate. (*Babcock and Wilcox Company.*)

heated by the flue gas and then cooled by the combustion air which in turn is heated. This type of preheater, it is claimed, occupies less space for the same heat recovery than other types.

621. In locating air preheaters care should be taken to obtain even gas distribution over the area of the heating elements. Dead pockets provide areas where the gas may cool below its dew point and where cinder may collect. Both factors cause corrosion of the preheater elements which are built for easy replacement because of the likelihood of corrosion. Sometimes arrangement is made to recirculate part

of the air to keep the gas temperature from falling below the dew point.

622. Temperature to which combustion air may be raised depends on the method of firing. With pulverized coal, oil, or gas higher temperatures are permissible, but with stoker the air preheat must generally be limited to a lower value to prevent burning of stoker parts. With stokers the air temperature ranges between 200 to 450°, with pulverized coal air up to 600 to 650° is used.

623. Air preheaters and economizers obviously decrease the draft that the stack can produce because of the lower temperature of the gas as it enters the stack. At the same time additional draft is required to overcome the friction resistance offered by either of these pieces of equipment. Friction resistance offered by either economizers or air preheaters varies between 2.5 to 4.0 in. water at the usual maximum output of the boiler. Because of this, it is almost always necessary to use induced draft when either or both of these heat-recovery apparatus are used. Where either equipment is justified, the operating cost of the fan is, ordinarily, much more than offset by the saving due to the economizer.

QUESTIONS ON DIVISION 23

1. In what manner does an economizer improve boiler-room economies?

2. Would you install an economizer in a noncondensing plant?

3. Why is a mechanical draft installed frequently in connection with economizers?

4. Why is an economizer more effective in absorbing waste heat than is additional boiler-heating surface?

5. Discuss the general considerations which affect the location of an economizer in relation to the boiler.

6. For what purpose is an air preheater used?

7. What are three types of air preheaters? Describe them.

8. What is the range of air temperatures usually used with stoker? With pulverized coal?

9. What precaution should be taken in locating air preheaters?

10. What causes corosion in air preheaters?

11. What resistance to gas flow is offered by economizers and air preheaters?

12. Why must induced draft fans be used when economizers and air preheaters are used?

DIVISION 24

FEED WATER AND FEED-WATER TREATMENT

624. Boiler feed water from natural sources is never pure. It will always be found contaminated with other substances in solution or suspension. The impurities may be either solids or gases. If the former, they are likely to be deposited in the boiler as the water evaporates into steam. If the latter, they will pass out from the feed-water heating equipment or with the steam.

625. The original or prime sources of feed-water supply for steam boilers may be broadly classified as follows: (1) cisterns of rainwater that may have drained from the roofs of buildings; (2) Lakes, streams, and reservoirs of surface water which may have drained from surrounding hills; (3) springs and wells of ground water which may have percolated through earth and rock strata.

NOTE.—*Rainwater* is invariably free from solid matter in solution. It may, however, be more or less contaminated with suspended impurities which are gathered from the air and from surfaces over which it flows. *Surface water* is, however, always contaminated more or less with dissolved solids and vegetable matter and perhaps with acids. *Well or spring water* is rarely free from soluble mineral matter.

626. The effects of impure water as boiler feed may be manifested in four different ways: (1) by foaming and priming, (2) by sludgy deposits on the water surfaces, (3) by scaly deposits on the water surfaces, (4) by internal corrosion (Sec. 633), affecting both the water- and steam-surfaces.

627. The impurities found in feed water from natural sources may be approximately classified as follows: (1) sedimental or sludge-forming substances, (2) scale-forming substances, (3) scum-forming substances, (4) corrosive substances.

NOTE.—*Sedimental or sludge-forming substances* in feed water usually consist of mineral and organic particles which are suspended in the water.

The mineral particles are formed of earthy or inert matter. The organic particles are formed of animal and vegetable matter. *Scale-forming substances* consist principally of lime and magnesia which are dissolved in the water in the form of carbonates, sulphates, and silicates. *Scum-forming substances* may be either mineral or organic. The mineral impurities may consist of soda in the form of a permanently soluble carbonate, sulphate, or chloride. The organic matter generally occurs in sewage-contaminated water. *Corrosive substances* may consist of chloride of magnesia, organic and mineral acids, or oxygen and carbon dioxide.

NOTE.—Water that contains incrustive ingredients, in excess of about 10 grains per gal. is known as *hard water*. It is *temporarily hard water* if the incrustive ingredients, as the carbonates of lime and magnesia are precipitated at a temperature of about 212°F. It is *permanently hard water* if the incrustive ingredients (such as sulphate of lime) are precipitated at temperatures above 300°F. But, of course, excessive concentration due to evaporation even at 212° will cause precipitation of solids.

NOTE.—Scale in steam boilers commonly betrays the presence of incrustive substances in the feed water. Such scale is an accumulation of impurities, which adhere to the water surfaces of the boiler in the form of a crust. It occurs in varying degrees of density and hardness. The scale-forming tendency of the incrustive substances may often be supplemented by that of sedimental substances. If the sludge, which is formed by sedimental deposits, lodges on the heating surfaces, it may unite with the incrustive precipitates and become baked to a hard rigid scale.

NOTE.—The most serious difficulty which attends the presence of scum-forming substances in boiler water is the foaming and priming which is likely to result. *Foaming* is a condition in which steam bubbles are enclosed in films of water of sufficient strength to hinder their coalescence and final bursting so that foam or a layer of bubbles is formed in the steam space of the boiler. It may occur where animal or vegetable oils in the feed water become saponified forming a sudsy scum. But a rational explanation of foaming is still unavailable. *Priming* is a condition in which large amounts of water are periodically carried out of the boiler with the steam. It may be caused by foaming, high water level, high steam or water velocity in the drum, or by a heavy glutinous skin on the surface of the water formed from an accumulation of a flocculent mineral matter in conjunction with organic matter.

628. The process by which soluble scale forming substances are precipitated from boiler water to form scale may, according to the nature of the substances, occur in either of two ways: (1) by the water losing, at the prevailing temperature, its power to hold the temporarily dissolved substances in solution; (2) by crystallization of the permanently soluble substances

through the excessive concentration which may result from continuous evaporation of the water.

Note.—The extent to which concentration of permanently soluble impurities in water in a boiler may proceed with safety, varies according to the nature of the impurities and the conditions of operation. It may range from 30 to 300 grains per gal. Concentration of chloride salts should not be over 500 p.p.m. expressed as chlorine and preferably as low as possible. The total solids in the boiler water are best kept below 1,700 p.p.m. Alkalinity which is dependent upon the silicates present should be between 100 to 250 p.p.m. The higher alkalinity is preferred when the silicate concentration is 100 to 200 p.p.m.

629. The characteristics of boiler scale vary according to the nature of the substance which forms the scale. Calcium and magnesia carbonates usually produce a soft porous scale which is more or less penetrable by water.

Oftentimes, however, the carbonate crystals are bound with fine particles of other material, giving a smooth uniform appearance. If a piece of carbonate scale is dropped into acid, carbon dioxide bubbles will come rapidly to the surface. Silica scale is the hardest type found. The scale is most often light colored, very brittle, and dense. It is not soluble in acid. Calcium sulphate produces an extremely hard and dense scale which is impervious to water. Magnesium sulphate alone produces a comparatively soft scale. But if both magnesium sulphate and calcium carbonate are present, they combine to form a particularly hard and flinty scale.

Note.—Magnesium carbonate may precipitate as a light flocculent scum-forming substance. If it encounters much grease while in this form, the two impurities may combine to produce a spongy scale which is exceptionally troublesome on account of its extremely high resistance to heat transfer.

Scale formed by organic matter—oil, sewage, trade wastes, or vegatable matter is usually dark colored, often brown. It is very light in weight and will often burn if ignited. It is usually soluble in strong nitric acid.

630. Four general methods are available for coping with impurities in boiler feed water: (1) The boiler may be opened periodically, the precipitated impurities loosened mechanically and the boiler washed out. (2) The acids may be neutralized

and the solids eliminated before the water is fed to the boiler. This is pretreatment of the water. (3) Substances may be injected into the boiler along with the feed water to minimize or obviate, by chemical or mechanical processes, deleterious action by the impurities. This is internal treatment of the water. (4) Only evaporated water may be used for feeding.

NOTE.—Unless the water contains none but noncorrosive sedimental impurities and these in very moderate quantities, the first method enumerated in the preceding section should not, in any case, be relied upon exclusively. The third method or internal treatment should be attempted only by prescription of a competent chemist. The second method or pretreatment is advantageous in any case. As a matter of economy, its use is generally imperative where the impurities are present in large quantities and are of complicated variety. The fourth method is used by central stations and when the percentage of make-up water is low. It is used in almost all cases where the boiler pressure is over 1,000 lb. Evaporators are treated in the author's book on "Power Plant Auxiliaries and Accessories."

631. Suspended Solids in the Water May Be Removed by Natural Settling or by Using Coagulants.—Natural settling requires an artificial pond of sufficient capacity to allow the water to remain practically quiescent about one day before use. The suspended solids settle out by gravity. Advantages include low cost and simplicity. Disadvantages include reduced effectiveness at low temperatures, space requirements usually necessitating outdoor location. In winter weather more or less difficulty may be expected in northern states. Settling of solids in a reasonable length of time requires a considerable water area. By using a coagulent, a floc is formed which has an affinity for the usual suspended solids. The coagulated matter settles to the bottom by gravity. Usual coagulants are alum or alum sulphate, copperas or ferrous sulphate. Addition of the coagulant greatly accelerates the settling action and hence reduces the space required for the settling pond or makes possible the use of inside tanks. Disadvantages include the cost of chemicals and desirability of supervision by a chemist. Improper control of coagulant proportions may result in increased hardness of the water

or in an increase in its corrosive properties. When coagulants are used it is usually necessary to use a suitable filter.

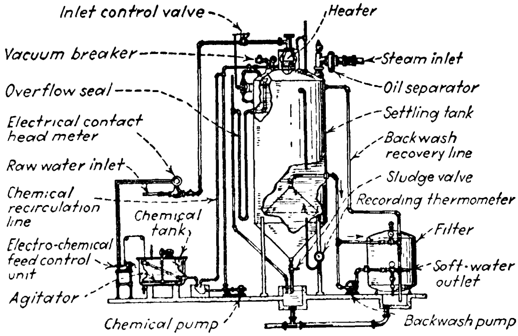


FIG. 435.—A continuous hot lime soda softner. (*Permutit Company.*)

632. Five general chemical processes of pretreatment of boiler water are available for scale prevention: (1) the lime process, (2) the soda process, (3) the lime-and-soda process

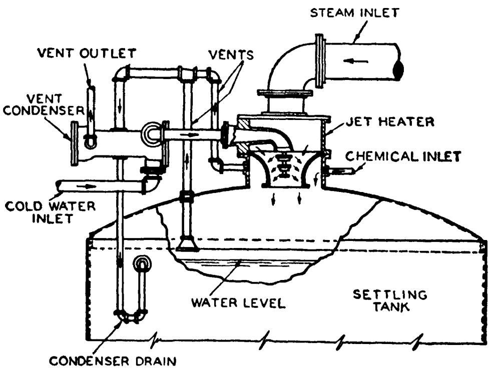


FIG. 436.—Single fixed jet-head heater for hot process softner. (*Permutit Company.*)

(Figs. 435 and 436), (4) the zeolite softners (Fig. 437), (5) the phosphate treatment. The lime process and the soda process are seldom applied separately.

NOTE.—The desirability of artificial purification of natural waters which are intended for boiler feed may be considered from two points of

view: (1) the relative impurity of water, (2) the expense of purification as compared with the expense due to the physical depreciation, the labor of cleaning, and the diminished evaporative efficiency of scaled, corroded, and sediment-containing boilers. Natural waters sufficiently pure for boiler feed are common in many localities. The impurities may be such that their deposits can be expelled from the boilers, practically as fast as formed, with blowoff apparatus. In these cases the boilers may be safely and profitably continued in service for periods extending from a few weeks to several months without being opened for cleaning. In other localities the untreated water available for boiler feed may be so loaded with impurities that the boilers can be run only two or three days between cleanings. The treatment necessary for purification should always be prescribed by a chemical expert.

633. The lime process of pretreatment of boiler water is used where the principal incrustive substances are calcium and magnesium carbonates. It is applied by introducing a quantity of milk of lime, which is a suspension of slaked lime.

634. The soda process of pretreatment of boiler water is used where calcium sulphate is the principal incrustive ingredients. It is applied by introducing soda ash (sodium carbonate).

635. Lime-and-soda process of softening water may be used when the feed water contains both carbonate and non-carbonate hardness. A sufficient amount of sodium carbonate is added to react with the sulphates of calcium and magnesium forming carbonates of calcium and magnesium. The calcium carbonate is unsoluble and precipitates, but the magnesium carbonate is soluble. Sufficient caustic lime, usually in the form of milk of lime, is added to react with the bicarbonates of calcium and magnesium forming calcium carbonate and magnesium hydroxide, both insoluble. In addition enough lime to react with the magnesium carbonate, resulting from the soda-ash reaction is supplied to form magnesium hydroxide and calcium carbonate, both insoluble. Caustic soda is used sometimes to precipitate magnesium sulphate but is not effective for calcium sulphate. It is somewhat preferable to soda ash for removing magnesium sulphate. Sodium aluminate is being used increasingly with lime and soda to remove calcium and magnesium. It has the advantage of forming a flocculent precipitate which settles out more quickly than

with other such precipitants and leaves soda ash, which is a softening agent. Barium salts (carbonate and hydrate) may be employed as softening agents and are preferable to sodium salts for removing noncarbonate hardness because the barium sulphate formed is insoluble whereas the sodium sulphate is soluble and concentrates in the boiler. But barium salts are expensive and hence are used only when trouble is experienced with high concentration of sodium sulphate in the boiler.

636. Chemical Softeners May Be Intermittent or Continuous.—Intermittent softeners operate on the fill-and-draw plan. A continuous softener, as the term indicates, softens the water as it flows through the apparatus. They may operate on either the hot or cold principle. Where there is a steady demand for water the continuous type is preferable unless storage is provided for soft water. As a rule, intermittent softeners permit more accurate dosing with chemicals, since the entire charge of the softening agent is added to the water at one time. Better settling is effected in intermittent then in continuous systems as there are no water currents. Continuous units, however, take less space as only one tank is required and are more flexible in operation.

637. An intermittent system of chemical feed-water treatment consists of two or more tanks, one supplying soft water while the other one is being filled, treated, and allowed to settle. The entire amount of chemical in proportion to the amount of carbonate and noncarbonate hardness is added to a tankful of raw water and the mixture agitated to insure thorough mixture of the chemicals and the water. The precipitate settles while the water is quiet and usually requires 8 to 10 hr. The soft water is then drawn off and sometimes passed through filters to clarify it.

638. A continuous softening system consists of a tank with baffling arrangement and compartments for chemical reaction and settling of precipitate, a device for feeding the chemical, and means for agitation. Chemicals are introduced as the water enters, usually at the top, in proportion to the rate of flow and the hardness. The water flows down in the reaction compartment where it is agitated to insure thorough

mixing with the chemicals and so increase the speed of reaction. It usually flows upward in the settling compartment. Upward velocity of the water in the settling compartment is not over 3 to 6 ft. per hr. Because of this velocity and the shorter time for settling, water from continuous softeners is usually filtered through nonsilicious material.

639. Cost of softening including capital charges varies from 2 to 10 cents per 1,000 gal. depending on type of equipment, nature of water, and completeness of softening required.

640. Hot-process chemical softeners are practically the same as cold-process softeners except that a heater is incorporated in the softener to heat the cold raw water to about 200° as it enters. The scale-forming solids are removed in part by the effect of heat and part as the result of chemical reaction. Velocity of reaction is more rapid in hot than in cold water and the insoluble salts formed settle more rapidly. Hot softeners can reduce the scale-forming solids to from 1 to 3 grains per gal. as compared to 3 to 5 grains per gal. for cold softeners and in addition eliminate some of the dissolved gases that may later cause corrosion. Figure 435 shows a hot-process softener comprising chemical tank and proportioner, reaction tank, settling chamber, heater, and pressure filters. A special feature is the uptake funnel through which the soft water passes. It is based on the principle that the heavier precipitates are thrown down as the direction of flow changes into the funnel and only the very fine particles are carried up with the soft water. These are so fine that it is preferable to filter them rather than provide means for them to settle.

641. Amounts of chemical required to soften water may be determined only on the basis of a raw water analysis. A simple method of estimating the required amount of softening reagent, recommended by the "Manual of American Water Works Practice," is as follows:

1. The sum of the free and half-bound carbon dioxide (expressed as parts per million) multiplied by 0.0106 = pounds of lime per 1,000 gal. necessary to absorb the free and half-bound carbon dioxide.

2. Total magnesium (expressed as parts per million) multiplied by 0.019 = pounds of lime per 1,000 gal. necessary to precipitate magnesium.

Soda ash required to remove noncarbonate hardness is found as follows: Noncarbonate hardness (expressed in parts per million) minus the quantity of noncarbonate hardness that is to be left in the feed water, multiplied by 0.009 equals pounds of soda ash required per 1,000 gal. It is customary to add a slight excess of the amount of soda ash theoretically required but it must be controlled carefully to avoid high concentration of this salt in the boiler.

642. Softening Water by Base-exchange Silicate.—Certain hydrous silicates called zeolites possess the property of removing calcium and magnesium from hard water passed through

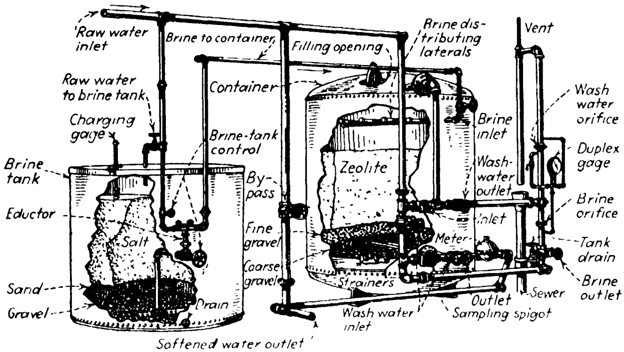


FIG. 437.—Diagram of zeolite system of water softening. (*Cochrane Corporation.*)

them and replacing these substances with sodium or potassium. The exchange takes place fairly rapidly so that water in passing through the mineral may be completely or almost completely softened. The zeolite material will continue to soften water until practically all the sodium has been exhausted from it. When this condition results the zeolite mineral may be regenerated, that is, have the sodium replaced that was removed during the softening process. Regeneration is accomplished by washing with a solution of common salt (sodium chloride), during which the reverse action takes place and the calcium and magnesium extracted from the water by the zeolite is passed into the brine solution and the sodium is taken up by the mineral. After this process the zeolite is washed with clean water to remove the salt. Before the

regenerating process the zeolite is usually backwashed to remove suspended solids that may have collected on the surface of the mineral. Zeolite softeners are usually installed in duplicate so that one may be regenerated while the other is in use. Figure 437 illustrates a cross section of a vertical zeolite softener and a zeolite system.

643. Zeolite softeners produce water that has from zero hardness to 1½ grains of hardness. It is essential that water to be softened by zeolite be free from suspended solids, and for this reason it is customary to provide a filter for clarification of the water prior to softening. The mineral acids and appreciable amounts of free carbon dioxide are objectionable and should be neutralized before the water is passed through the zeolite.

644. Advantages and Disadvantages of Zeolite Softeners. *Disadvantages.*—When the hardness of water is due to bicarbonate only, lime alone is required to soften it, and the process will cost not more than perhaps half the cost of pretreatment by zeolite system. Water that contains high amounts of sodium salts, in addition to calcium and magnesium, may be softened more satisfactorily by chemical softeners than by zeolite systems. The zeolite softener cannot be operated economically with waters of high hardness. They increase the sodium salt solution in proportion to the lime and magnesia salts removed from the raw water supplied, and consequently there is a tendency of boiler water to have high concentration of soda salt, unless boiler blowdown is increased. There is some loss of zeolite in operation.

Advantages.—Practically complete removal of lime and magnesia salts from water. No chemicals are added to the water. The softener may be placed on the main supply line and will cause practically no loss of pressure. No appreciable amount of scale will deposit in the boilers. The system will operate efficiently with fluctuating hardness of the untreated water. It may be used to deliver water of practically zero hardness when operating on a very hard water, if the water is treated first by lime and soda softener.

645. Embrittlement of boiler metal and resulting intercrystalline cracking has been a serious condition occasionally

found in riveted boilers. It is claimed by many feed-water experts that this condition is due to high concentration of caustic soda in the presence of silicate and highly stressed metal and relatively low concentration of sodium sulphate. The concentration of caustic soda necessary to produce embrittlement does not normally exist in the boiler water as a whole, but it is claimed that sufficiently high concentrations develop in the joints of riveted shells and where tubes are rolled in to cause this effect. Certain chemicals appear to inhibit the embrittling effect of caustic soda and the A.S.M.E. "Suggested Rules for Care of Power Boilers" recommends the use of sodium sulphate for this purpose. The ratio of sodium sulphate to total alkalinity expressed as sodium carbonate given in the code is as follows and should be maintained at not less than these values.

Boiler Pressure Lb. per Sq. In.	Ratio Sodium Sulphate to Total Alkalinity Expressed as Sodium Carbonate
Up to 150	1
150 to 250	2
Above 250	3

To maintain this sulphate-carbonate ratio sodium sulphate may be added either with the chemicals in a hot-process softener or after the softening process. Sulphuric acid may also be added to maintain the sulphate-carbonate ratio, but its use should be under the direction of a competent chemist.

646. Phosphate Treatment.—Various forms of phosphate, such as disodium phosphate and trisodium phosphate, may be used in feed-water treatment. It is recommended for pressures above 250 lb. that sodium phosphate be used to inhibit scale formation. Addition of sodium phosphate prevents the formation of troublesome silicious scales which sometimes occur when the feed water contains silica together with calcium or magnesium. Treatment with sodium phosphate is usually made supplementary to other water softening processes and generally removes the last trace of calcium carbonate throwing down a fine nonsticking sludge. Phosphate solution may be injected into the boiler drum or into

the feed line ahead of the boiler. Some authorities believe it tends to inhibit embrittlement, having much the same effect as sodium sulphate.

647. Pretreatment for counteracting the destructive effects of corrosive substances in feed water involves both mechanical and chemical processes. When the corrosive substances consist of oils, and other organic matter which distill acids, their removal may usually be effected by filtration. The corrosive effect of oxygen may be obviated by expelling the air from the water in deaerating heaters and subsequent treatment with sodium sulphite. It is extremely important that oxygen be removed from the feed water especially when economizers are installed and with high pressures. Preferably oxygen should be reduced to zero but not over 0.05 c.c. per liter should be allowed. Other corrosive elements respond only to chemical pretreatment. pH of boiler water should be maintained at not less than 10.5.

NOTE.—Mineral acids rarely occur in appreciable quantities in waters from natural sources. When present, however, they may be neutralized readily by the lime which is used in the processes of pretreatment for incrustive impurities. The corrosive property of chloride of magnesia is eliminated by precipitation of the magnesia in the soda process of pretreatment. Dissolved carbonic acid is also neutralized in the lime-and-soda process.

NOTE.—pH is the measure of acidity or alkalinity of the water. A pH of 7 indicates a neutral water. A pH below 7 indicates an acid water and above 7 indicates an alkaline water. A change of one pH indicates a change in the degree of acidity or alkalinity of 10 times the original. pH may be determined by indicator solutions and color comparison.

648. Crude oil and kerosene as scale preventives are comparatively ineffective. Serious trouble may attend their use as such.

649. Boiler compounds or scale solvents generally contain soda and tannin. In some cases they contain vegetable ingredients which are presumed to envelope the particles of incrustive substances with a film or skin which prevents their adherence to the metal surfaces. They are generally fed directly into the boiler drum. Some of the many boiler compounds available produce the results claimed for them but others do not, and in general it is safer to avoid their use

unless advised by a chemist familiar with boiler feed-water treatment problems.

NOTE.—The purpose of soda in boiler compounds is to reduce the hard scale to a sludgy mass. The purpose of tannin is to penetrate between the scale and the boiler metal. The soda is employed to reduce the scale to a form in which it may be readily blown out. The tannin is intended to simplify removal of the scale when the boiler is opened for cleaning. If the use of a scale solvent is necessary, it should be introduced in small quantities at frequent intervals through an apparatus which may be attached to the feed pipe.

650. Chemical treatment of feed-water to prevent scale, corrosion, and embrittlement is very complex and becomes increasingly difficult when the boiler pressure becomes high. The previous sections explain briefly only the processes of treatment now in general use. They should not be applied without an analysis of the feed water and the advise of a chemist thoroughly familiar with the problems of boiler feed water treatment.

QUESTIONS ON DIVISION 24

1. What becomes of gaseous impurities in feed water?

2. What are the most conspicuous evidences of impure feed water?

3. How is sediment formed in a boiler?

4. How is scum formed?

5. In what state or condition do incrustive substances dwell in feed water?

6. What is temporarily hard water? Permanently hard water?

7. What are the principal incrustive ingredients in boiler water?

8. What is boiler scale?

9. What are the principal corrosion impurities in feed waters?

10. How may sludgy sedimental deposits develop into scale formations?

11. What phenomena mainly indicate the presence of scum on the surface of boiler water?

12. What is the nature of the scum that ordinarily causes foaming? That causes priming?

13. By which natural processes does precipitation of incrustive impurities in boiler water occur?

14. What is the safe maximum degree of concentration of soluble or dissolved impurities in boiler water?

15. How may excessive concentration be prevented?

16. What is the chief physical characteristic of the scale which is formed by the carbonates of lime and magnesia? Of scale which is formed by the sulphates of lime and magnesia?

17. What four distinct procedures are available for coping with the impurities in feed waters?

18. Upon what considerations does the desirability of artificial purification of feed water mainly rest?

19. Describe the lime process of purification. The soda process. The lime-and-soda process.

20. What are the active principles of boiler compounds?

21. What are prime sources of feed-water supply?

22. What sort of impurities may be found in rain water? In surface water? In spring water?

23. Describe the intermittent system of feed-water treatment.

24. Describe the continuous system of feed-water treatment.

25. What are zeolites? What property do they possess?

26. Describe the zeolite system of water softening. When is it applicable?

27. What causes embrittlement of boiler metal?

28. How may embrittlement be prevented?

29. For what purpose is the treatment with phosphate?

DIVISION 25

MANAGEMENT, INSPECTION, AND MAINTENANCE OF STEAM BOILERS

651. Management of a boiler embraces such activities as are necessary to continuous and efficient operation. These activities are directed mainly as follows: (1) to insure, at all times, a full supply of the best quality of feed-water obtainable; (2) to provide an adequate supply of fuel and to burn it economically; (3) to furnish a continuous supply of steam, at a constant pressure, for all of the purposes for which the plant is designed.

652. Inspection of boilers comprises such activities as are necessary to the acquisition of precise knowledge regarding the physical condition of all parts of the boilers and their accessories.

NOTE.—Inspection of the furnaces, settings, and chimneys of boiler plants is correlated with inspection of the boilers themselves.

653. The maintenance of a boiler comprises certain duties which are necessary to preservation of the working efficiencies of the boiler and its appurtenances. The appurtenances include furnaces, settings, fans, chimneys and other accessory apparatus. The duties of maintenance tend to conserve both economy and safety.

654. Control of the feed-water supply should receive vigilant attention. The water should be fed to the boiler continuously. With constant load the quantity flowing in should be as nearly equal as practicable to the quantity going out in the form of steam. By observing this rule, the highest feed-water temperature possible with the available heating apparatus is fully realized. Likewise, a continuous supply of steam, as dry as the structural limitations of the boiler will permit, is assured.

655. Dangerously low water in a boiler is an emergency wherein a portion of the heating surface is without the protection of water contact. It occurs rarely under proper management. Nevertheless it is a condition that should be anticipated.

NOTE.—When the water becomes dangerously low the boiler should be cooled as quickly and as safely as possible. If the feeding apparatus is already working, it should be allowed to continue. Otherwise it should not be started. Neither should it be speeded up. Usually, the fire may be most conveniently smothered by covering it with wet ashes. All doors and passages through which cooling currents of air may pass, to the combustion chamber, should be opened. When it is certain that no water is being evaporated, the fire may be drawn and the boiler emptied. The boiler should not be continued in service until the condition of the exposed surface has been determined by competent inspection. If the boiler is fitted with fusible plugs, the ends of the plugs should be kept free from insulating deposits of soot and scale. This is vital to their prompt action (Fig. 438) in the event of dangerously low water.

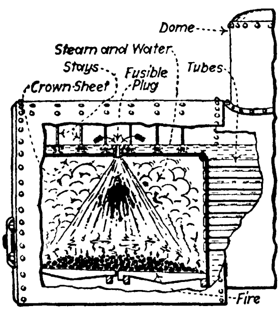


FIG. 438.—Fusible plug giving notice of low water in a horizontal fire-box boiler.

656. Economy of Boiler Operation Is Mainly a Matter of Skillful Firing.—Efficient firing is in evidence when the fires are kept at an even thickness, when holes are not permitted to burn in the fuel bed, and when ash and clinker are as carefully removed from the sides and corners of the furnace as from the middle surfaces. The depth of the fire should be adjusted in accordance with the demand for steam. The draft should be so controlled that it will produce the most economical percentage of CO_2 (see Div. 16, Combustion and Firing). The cleaning of the fires should be attended to systematically.

With pulverized-coal and oil fuel, smoky or sparking flame should be avoided and flame travel should be adjusted so as not to impinge against the furnace walls.

NOTE.—When ash and clinker have accumulated to the extent of darkening the ashpit, cleaning of the furnace should not be delayed.

657. Proper Use Should Be Made of the Blowoff Apparatus. In the average boiler plant, which uses ordinary feed water, one of the effects of evaporation is a constantly augmented concentration of the impurities in the boilers. This may be avoided by blowing down frequently. Formation of scale may also be diminished by frequent use of the blowoff apparatus.

NOTE.—In a test of a few day's duration without blowing off, the foreign matter in the water in a boiler increased from 25 to 750 grains per gal. It required but a comparatively short time, when the water was not changed by blowing off, to produce a solution containing over 1,000 grains to the gallon. There is a marked difference in the quantity of heat absorbed under this condition, which is unfavorable alike to efficiency and longevity of the boiler.

NOTE.—Most of the suspended solid matter in boiler water is precipitated as a loose sediment. Scaly deposits, on the other hand, are usually due to mineral matter which is held in solution.

658. The proper time for blowing scum and sediment from a boiler is when the water is quiescent. This occurs in the intervals when there is practically no outflow of steam, as at noontime in a manufacturing plant, or early in the morning, before the plant is in operation. Blowing off while the water is quiescent insures that the sediment- and scum-forming substances have had time to so accumulate as to be carried out with the currents of water issuing through the blowoff piping.

659. The proper procedure in handling the bottom blowoff values when blowing out an accumulation of sediment is this: (1) Open the *auxiliary valve* or cock C (Fig. 439). (2) Open the *blowoff valve* V. (3) When the water level has been blown down to a depth of about 2 or 3 in., close V. (4) Close C. (5) V should again be opened for a short interval to permit the water in it to drain out, after which it should be closed finally.

NOTE.—Two blowoff valves (C and V, Fig. 439), connected in series, are specified for power boilers by the A.S.M.E. Code. One valve provides insurance against failure of the other. In practice, the *blowoff valve* V is specially designed to withstand the excessive wear and tear of blowoff service. This wear is caused by the rushing currents of sediment-charged water. The *auxiliary valve* C may be a relatively low-priced

and simple cock. By operating the valves in the sequence specified above, the erosion, due to wire drawing when the valve is almost closed, is imposed on the specially designed valve *V*.

NOTE.—A leaking blowoff is a source of serious loss. It has been computed that, when a boiler carries a working pressure of 150 lb. per sq. in., 36,000 gal. of water will be wasted in a month's time through a leak in the blowoff corresponding to a circular orifice 0.0625 in. in diameter. With coal having a heating value of 13,000 B.t.u. per lb., and costing $3.00 per ton, and water metered at the rate of 10 cents per 1,000 gal.,

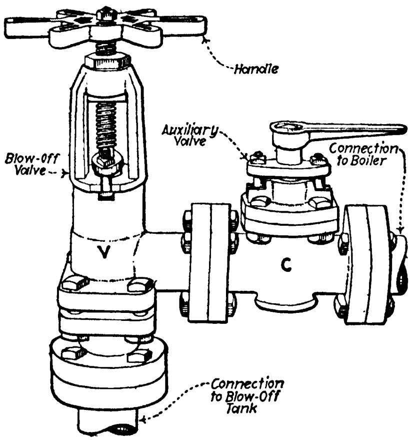


FIG. 439.—Proper arrangement of blowoff valves.

the loss due to lost water and lost heat would amount to over $12.00 per month of boiler operation.

660. Blowing down the water column is vital to the safety of the boiler and to the durability of the water-column connections. The quiescent condition of the water in the column invites clogging of the connections by mud-forming impurities in the water. The column should in any case, be thoroughly blown down at least twice a day. Where the water is more than ordinarily impure, more frequent blowing, perhaps as often as four or five times on a watch, might be necessary.

661. The condition of the top and the bottom connections of the glass water gage is indicated by the behavior of the water when returning to the normal height in the glass after the glass has been blown out.

NOTE.—If the water rushes instantly to the top of the glass, and then recedes slowly to the normal level, it is a sure sign of a restricted steam passage. If the water travels and settles slowly and deliberately, up the glass and settles quietly at a plane coincident with the true level, then an obstruction undoubtedly restricts the water passage. But if the water shoots up the glass with high velocity, its momentum carrying it a little above the regular stage, at which it settles after a few rapid fluctuations, the action indicates perfect freedom in both passages.

662. The care exercised in cleaning a boiler (Fig. 440) largely affects its safety and economy. The quality of the

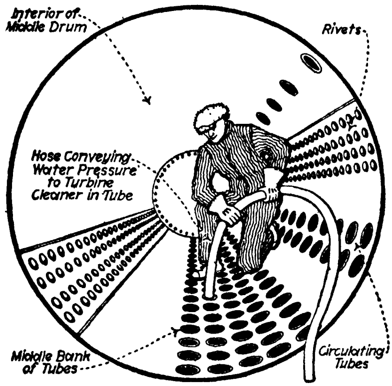


FIG. 440.—Cleaning the tubes of a Stirling boiler.

feed water mainly determines the interval that should elapse between successive internal cleanings. The kind of fuel and the quantity burned are the principal factors in regulating the periods of external cleaning. However, it is, in any case, essential to economy that soot deposits be blown (see Div. 14 for Soot Blowers) from the tubes at least once a day. When the boiler is laid up for internal cleaning, all manhole and handhole plates should be removed so as to provide maximum accessibility to all parts.

663. Cleaning the interior of a new boiler, before the boiler is placed in service, is very necessary. The plates of a boiler acquire a coating of oil while they are being assembled. Foaming will inevitably result if this is permitted to remain. The interior surface may be cleaned by scouring it with an

alkaline solution. Water surfaces of new boilers are often coated with corrosion-resistant paint, such as "Apexior."

NOTE.—A quantity of soda ash should be placed in the boiler. The boiler should be filled with water to the top gage cock. A fire should be started and maintained slowly for about 12 hr. The boiler should then be allowed to cool slowly. When the water has become tepid, the boiler should be emptied and washed out. The soda ash dissolved in the simmering water will have cut the greasy coating from the plates and tubes. Therefore, the grease will readily pass out with the water. About 1 lb. of soda ash should be used for each 15 cu. ft. or 940 lb. or 110 gal. of water which is put into the boiler.

664. Cutting a boiler into the line consists in opening the connection between the boiler and the main header or steam

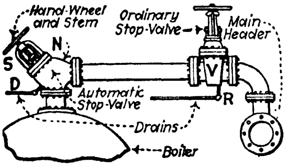


FIG. 441.—Steam connection equipped with automatic and ordinary stop valves.

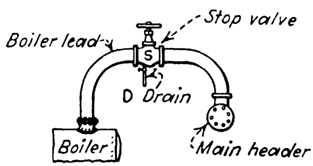


FIG. 442.—Steam connection equipped with single ordinary stop valve.

line. It is the operation of placing the boiler in service. In a boiler plant composed of two or more units, this operation usually occurs while other boilers are already delivering steam to the main header. It may then be done automatically if the boiler is equipped (Fig. 441) with an automatic nonreturn stop valve. Otherwise (Fig. 442) it must be done by manipulation of an ordinary stop valve.

NOTE.—When a fire is started under a boiler which is equipped (Fig. 441) with an automatic nonreturn stop valve N, the operator should first see that the drains D and R are open. He should next open slowly the stop valve V and close the drains D and R. He may then screw the stem S of the nonreturn valve N out to the wide-open position. Until the steam in the boiler attains the same pressure as that in the main header, the disk of the nonreturn valve will be seated by the force of its own weight alone. The disk will, therefore, rise automatically from its seat in response to a further very slight increment of pressure per square inch in the boiler.

When a fire is started under a boiler which is not provided (Fig. 442) with an automatic nonreturn stop valve, the operator should see that

the drain D is open. When steam begins to blow strongly through it, D should be closed. When the pressure in the boiler becomes equal to or slightly less than the pressure in the main header, the stop valve S should be opened very cautiously for about two turns. It may then be opened up rapidly. Extreme care should be taken to guard against opening the

SUMMARY OF DAILY BOILER-ROOM LOG FOR WEEK ENDING Nov. 15 - 19

DAY	BOILER NO. 1					BOILER NO. 2					BOILER NO. 3					BOILER NO. 4					AVERAGE FLUE TEMPERATURE DEGREES FAHR.	AV. FEEDWATER TEMPERATURE DEGREES FAHR.	ASHES REMOVED POUNDS
	COAL BURNED, POUNDS	WATER EVAPORATED POUNDS	AV. STEAM PRESSURE LB. PER SQ. IN.	AVERAGE DRAFT, INCHES	AVERAGE CO2, PER CENT.	COAL BURNED, POUNDS	WATER EVAPORATED POUNDS	AV. STEAM PRESSURE, LB. PER SQ. IN.	AVERAGE DRAFT, INCHES	AVERAGE CO2, PER CENT.	COAL BURNED, POUNDS	WATER EVAPORATED POUNDS	AV. STEAM PRESSURE, LB. PER SQ. IN.	AVERAGE DRAFT, INCHES	AVERAGE CO2, PER CENT.	COAL BURNED, POUNDS	WATER EVAPORATED POUNDS	AV. STEAM PRESSURE LB. PER SQ. IN.	AVERAGE DRAFT, INCHES	AVERAGE CO2, PER CENT.			
SUN	21050	215400	150	07	110						540	4900	—	—	—	490	5700	—	—	—	310	210	1800
MON	19600	19570	150	072	105	Out					20500	185900	150	072	95	4600	22200	150	072	11.5	330	208	5500
TUES	18600	180600	150	07	110		of						150	07	95	21500	216000	150	07	11.5	320	207	5100
WED	21700	201400	150	069	115			Service					150	069	100	14600	140000	150	069	100	310	210	4400
THUR	20600	207100	150	07	10.00								150	07	105	16000	151000	150	07	100	340	209	4300
FRI	18950	174300	150	071	95								150	071	110	20100	200000	150	071	95	310	209	5600
SAT	19870	199100	150	071	100								150	071	110	15100	156000	150	071	100	315	210	4100

RECORD OF CLEANINGS AND REPAIRS OF BOILERS, FURNACES AND ACCESSORIES

BOILER NO. 1 Glass water gage renewed.

BOILER NO. 2 Drums washed out and tubes turbined. Two defective tubes (4th and 8th in bottom row) replaced with new ones. Outer blow off valve ground in. Inner blow-off valve repacked. Fusible plugs renewed.

BOILER NO. 3 Diaphragm of damper regulator renewed.

BOILER NO. 4 Disc of feedwater regulating valve renewed.

RECORD OF CONDITION OF BOILERS, FURNACES AND ACCESSORIES

BOILER NO. 1 Furnace arch needs renewal.

BOILER NO. 2 Placed in serviceable condition during week.

BOILER NO. 3 Check valve leaks. Automatic stop valve needs overhauling.

BOILER NO. 4 Safety valve leaking slightly. Head of one stay bolt in rear water leg needs calking.

R E M A R K S

No. 3 and 4 boilers banked from 2:00 A.M. Sunday to 3:00 A.M. Monday.
Quality of coal delivered during week was very irregular.
Base of chimney needs cleaning.

(SIGNED) William Jones
CHIEF ENGINEER

FIG. 443.—Suggested record form for boiler-room use.

stop valve S before the pressures have become equalized, or to delay its opening until the boiler pressure exceeds the header pressure. Proper performance of this duty mainly depends upon the accuracy of the steam gages on the boilers.

665. A boiler-room log (Fig. 443) is indispensable in maintaining a proper standard of operating efficiency. It is

also a valuable guide when plans for improvement are being devised. It should exhibit a record of all important data pertaining to the condition and operation of the boiler plant.

NOTE.—The basis of an adequate system of records in a boiler plant is the daily log, a weekly summary of which is suggested in Fig. 443. The average values which are given in the summary are presumed to be the daily averages of values which have been read hourly on the indicating instruments and recorded in the daily log.

When a boiler plant is equipped with a recording thermometer in the feed-water line, a recording pyrometer in the chimney connection, a recording draft gage, a recording steam gage, and a recording flue-gas analyzer, the charted records which are taken from the instruments afford, in themselves, a connected history of the plant's performance.

For descriptions of steam-power-plant recording instruments, see the Author's "Practical Boiler-room Economy" and his "Practical Heat."

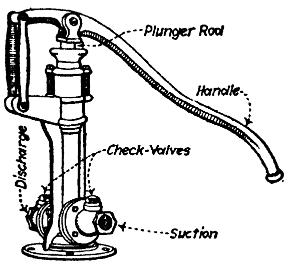


FIG. 444.—A hydrostatic test pump. (*Walworth Manufacturing Company.*)

666. Inspections of Boilers Should Be Made Systematically. At intervals of at least three months, each boiler in a plant should be examined critically by the engineer in charge. The sheets, seams, and tubes should be investigated for evidences of distortion, fractures, and corrosion. The braces and stays should be tested to determine their soundness. The search for defects should also include every detail of the accessory apparatus and of the furnace and setting.

667. Tests for the discovery of defects in a boiler are as follows: (1) the hammer test, (2) the hydrostatic test. A hammer test consists in tapping all parts of the boiler with light hammer blows. The condition of a part so tested is revealed both by the sound which is emitted and by the quality of the vibration which is set up in the fiber of the metal. The vibration is noted by applying the finger tips to the metal close to where the hammer blow is being struck. A hydrostatic test consists in filling the boiler with water, attaching a small specially designed hand pump (Fig. 444) to a convenient

opening in the boiler, and then pumping up a hydraulic pressure about 50 per cent greater than the steam pressure which the boiler is required to carry. (See A.S.M.E. boiler Code.)

NOTE.—The hammer test should always be used to determine the condition of a boiler. The hydrostatic test alone is not to be relied upon for revealing dangerous conditions. Hydraulic pressure is valuable chiefly in determining the tightness of joints and riveted seams.

668. Evidences of corrosion on the plates and seams of a boiler should be looked for constantly and diligently. They may be found both externally and internally.

669. External corrosion of boiler surfaces is generally produced by leaks through riveted seams and by drippings from the joints of piping and other fittings. When moisture spreads between the sheets of a boiler and the surrounding brickwork, it cannot escape readily by evaporation. Corrosion of the metal will therefore result.

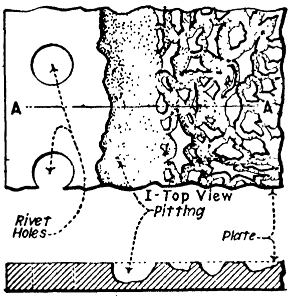


FIG. 445.—Pitting of a boiler plate.

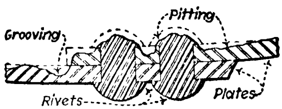


FIG. 446.—Grooving of plate and pitting of plate and rivets.

Moist ashes in contact with the boiler metal is also a common cause of external corrosion. The sulphur in the coal distills, during combustion, forming an acid gas. Deposits of soot in the passages through the boiler will absorb this gas. If the gas-saturated soot becomes moistened by water from a leak or other source, the gas will be converted to sulphuric acid. This is particularly ruinous to boiler metal.

670. Corrosion of the interior surfaces of boilers may be manifested as follows: (1) by uniform wasting of the plates and rivets, (2) by pitting or honeycombing (Fig. 445), (3) by grooving or channelling (Figs. 446, 447, 448, 449 and 450). In all cases, the principal cause is the same. This, stated generally, is chemical reaction between the boiler metal and

an acidulated and air-impregnated body of water. Internal corrosion may also be due occasionally to galvanic action.

671. Uniform wasting of boiler metal may be difficult to detect. It is a very insidious form of corrosion. This is due to the evenness of its progress throughout very extended areas of the plate. The hammer test will generally reveal its presence to the skilled boiler inspector. If any doubt should exist as to its progress, a 0.5-in. hole should be drilled in the suspected area so that the plate thickness may be calipered. The hole may then be threaded and plugged.

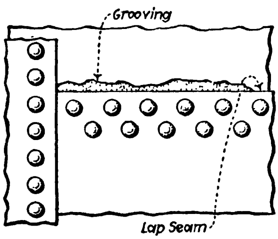


FIG. 447.—Grooving at edge of lapped plate.

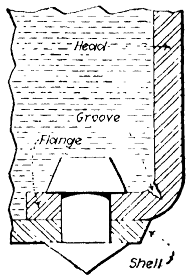


FIG. 448.—Groove corroded in fillet of head flange.

672. Pitting or honeycombing (Fig. 445) may, as a rule, be observed readily. It appears as more or less extended aggregations of irregularly shaped depressions in the metal. It may be found in all parts of the interior surface of a boiler, sometimes above but usually below the water line. The pittings are frequently found filled with a powdered substance composed of iron and mineral and organic matter precipitated from the water. This is frequently due to oxygen in the feed water.

673. Grooving or channelling usually occurs along the edges of lapped seams (Figs. 446 and 447), the rounded corners of flanges (Figs. 448 and 449), the edges of mud rings (450), and similar locations. It is often difficult to detect. It may originate due to inherent local rigidity of a certain part of the boiler metal which prevents the part from conforming readily to alterations in form induced by expansion and contraction. This resistance of a part of the metal to change of

shape causes the adjacent metal to bend a correspondingly greater amount. The increased bending of the adjacent metal may be sufficient to continually disrupt formations of scale or rust along a narrow length of its surface. Thus a strip of clean metal is being constantly exposed to the attack of the acids in the water and the subsequent corrosion. Grooving is the result. The *breathing* action of a boiler which is due to

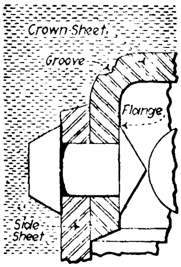


FIG. 449.—Groove corroded in outer curve of flange of crown sheet in horizontal fire-box boiler.

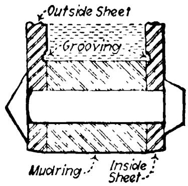


FIG. 450.—Grooves corroded, at edges of mud-ring, in sheets of horizontal fire-box boiler.

variations of pressure, may bend parts of the shell sufficiently to cause grooving.

674. When operation of a boiler plant is to be discontinued for an extended period, special precautions should be adopted to guard against deterioration. The boilers should be cleaned thoroughly, both internally and externally. All deposits of soot and ash should be scraped, brushed, and blown from the plates and tubes. Scale should be removed from the interior surfaces. The tube ends and riveted joints should be examined for leaks. All defective joints should be repaired. The surfaces, both inside and outside, should then be so treated as to ward off corrosion.

NOTE.—Corrosion in boilers which are out of service may be prevented in several ways. If the period of idleness is to be less than about three months, the boilers may be kept full of water. About 1 lb. of soda ash for each 15 cu. ft. of space should be dumped into the water in each boiler. Slow fires should be started. The water should be heated to a steaming temperature in order to expel the air from the boilers. The fires should then be drawn and the boilers pumped full.

If suspension of service is to continue longer than about three months, the boilers should be emptied, cleaned, and dried thoroughly. The

surfaces, both inside and outside, may then be coated with crude mineral oil. Or crude oil may be used on the inner surfaces and boiled linseed oil on the outer. Linseed oil will form a more durable coating than will crude oil. Otherwise the exterior surfaces may be painted with either red lead or tar. Portions of the exterior surface which are beyond convenient reach may be given a protective coating by the burning of tar beneath them. The tar thus volatilized will float upward to the otherwise inaccessible surfaces and cling thereto.

A film of crude oil may be spread upon the interior surfaces by filling each boiler with water, adding about 10 gallons of the oil, and then opening the blowoff valves. As the water level descends, the floating oil will adhere to the plates and tubes. A boiler so treated must be purged with soda ash (Sec. 663) before it is again placed in service. If pitting is noted in the drums of water-tube boilers, a coat of zinc paint should be applied.

The smoke stacks and chimneys of idle boiler plants should be covered with watertight hoods.

QUESTIONS ON DIVISION 25

1. What activities in a boiler plant fall within the meaning of the word *management?* Of the word *inspection?* Of the word *maintenance?*

2. What visible criterion of proper control of the feed-water supply is most apparent?

3. What benefits are realized by close regulation of the feed to a boiler?

4. What is the proper course of action in a low-water emergency?

5. What are the visible evidences of skilful firing?

6. What benefits result from proper use of the blowoff facilities?

7. When is the best time to blowdown a boiler? Why?

8. In what order should the two valves in a blowoff connection be opened and closed? Why?

9. When the glass water gage is blown down, what symptoms indicate the condition of the passages in the connections?

10. What considerations should determine the frequency of the internal cleanings of a boiler?

11. How may a new boiler be cleaned of oil?

12. What manipulation is necessary for cutting in a boiler which is furnished with an automatic nonreturn stop valve?

13. What should be the procedure in cutting in a boiler? Why?

14. What is the fundamental reason for maintaining a boiler-room log?

15. What is a hammer test? A hydrostatic test?

16. Why is damp soot a prolific source of corrosive activity?

17. How may the depth of penetration of uniform wasting in a boiler plate be known exactly?

18. In what parts of a boiler does pitting occur?

19. What is the cause of grooving?

20. How should a boiler be laid up for a temporary period? For a prolonged period?

DIVISION 26

SELECTION OF STEAM BOILERS

675. The selection of steam boilers is a subject which naturally classifies into two divisions: (1) selection as to type, (2) selection as to size or rating. So many different factors affect the situation that it is possible to submit only general suggestions. Each project should be considered individually and treated on its merits.

676. Some factors which should be considered when selecting the type of boiler for a given service are as follows: (1) character of load, whether steady or varying, duration and magnitude of peaks, plains, and valleys in the load graph and load factor; (2) space available for installation; (3) labor available for operation; (4) can boiler be shut down for cleaning? (5) Feed water available; (6) draft pressure which is available; (7) will or will not mechanical stokers be used? (8) efficiency of boiler; (9) kinds of fuel available; (10) initial and maintenance costs.

677. The Determination of the Most Economical Type of Boiler for a Given Installation Cannot Be Based on Any Single Feature.—On the contrary, all possible factors should be given consideration. The problem usually resolves itself into one of determining what type of boiler or boilers will generate, under the conditions obtaining in the plant in question, a pound of steam for the least money. Or, assuming that the same amount of steam will be generated in a year by each of the different tentative boiler installations, the problem is then one of determining which of the installations will have the least annual cost. Only the general method of procedure will be outlined. The detail process will vary for different cases.

Explanation.—First, determine the total first costs of complete installations of boilers of the different types which are under consideration. Each of the different installations should be designed to generate

the same number of pounds of steam per hour which is required by the project. The first costs of the different tentative installations having been thus determined, ascertain the total annual cost for each. The *annual cost A* will (Table XXII) be the sum of the items $D + I + N + T + R + C + H + F + M$. The installation which has the least annual cost will be the most economical.

TABLE XXII.—METHOD OF DETERMINING THE ANNUAL COST OF STEAM-BOILER OPERATION

Item	How computed	Total costs
Fixed Charges		
Depreciation	Amount laid aside yearly to replace unit when it is worn out. Based on first cost and life of equipment. Ranges from 20% of first cost each year for a 5-year life to 4% for a 25-year life	D
Interest	Cost of interest per year on the money which is invested in the boiler installation. Figure at prevailing rate. Usually 6 or 7% of first cost, per year	I
Insurance	Yearly cost of insuring the installation against fire, explosion, etc. Figure at prevailing rate usually about 0.75% of first cost, per year	N
Taxes	Yearly cost of taxes on the installation. Figure at prevailing rate, usually about 0.75% of first cost	T
Rent	Rental cost yearly of floor space occupied by boiler	R
Operating Expense		
Coal	Cost of the coal burned in a year	C
Coal handling	Yearly cost of labor and other expense incurred in handling the coal after it reaches the plant	H
Firing	Yearly cost of firing the coal	F
Maintenance	Yearly cost of repairs, labor, and material necessary to maintain plant in operating condition.	M
Total annual cost,	$A = D + I + N + T + R + C + H + F + M =$	A

678. The selection of a return-tubular vs. a water-tube boiler may usually be determined on this basis: For units with

between say 750 and 1,500 to 1,750 sq. ft. of heating surface, a return-tubular boiler is usually, everything considered, more economical than a water-tube boiler. While the water-tube boiler may be somewhat more efficient when operated under ideal conditions, its higher first cost, in the smaller capacities per square foot of heating surface, ordinarily, where these smaller capacities are concerned, more than offsets the possible increase in economy that may occur through its use. Not only do the small water-tube boilers themselves cost more per square foot than do the return-tubular boilers, but they also cost more to install. The cost per square foot of small water-tube boilers is much greater than the cost of the large ones. In fact, for the reasons just outlined, water-tube boilers are seldom manufactured in capacities smaller than 1000 sq. ft. On the other hand, return-tubular boilers are seldom made in capacities greater than 1,750 sq. ft. because if so made they would be altogether too bulky for practical and economical handling.

679. In selecting the preferable type of water-tube boiler for a given set of conditions, the factors cited in Sec. 675, Table XXII and possibly others should be given consideration. In general, all of the different makes and designs of water-tube boilers which are made by reputable manufacturers will, when operated at their rated capacities, show practically the same economies. Some are, however, more expensive to maintain than others. Again, some respond more readily to forcing for overloads than do others. Furthermore, some may operate more economically at certain overloads than do others. Reliable information as to comparative maintenance costs is difficult to obtain and must usually be based on one's personal experience or on the personal experiences of one's acquaintances. The boiler manufacturers can usually give efficiency guarantees for normal and overload operation.

680. In selecting a boiler which will be subjected to heavy overloads, on which it should operate with high economy, a water-tube boiler should be chosen which (1) presents to the combustion gas the maximum surface for heat transfer, (2) has large steam-liberating surface, (3) has unrestricted water circulation.

681. In new installations width and depth of the boiler are best chosen to meet the dimensions of the furnace and firing equipment which should be large enough to burn efficiently the fuel at the required rate and with economy in maintenance. It often happens that a premature decision as to location of building and boiler-supporting steel limits the free choice of boiler width, depth, and height, adversely affecting both first cost and efficiency.

682. Economy of first cost often acts in opposition to the dimensions required by considerations of over-all boiler efficiency. For example, a straight-tube, cross-drum boiler costs less when the tubes are long and the boiler narrow because headers and long drums are costly. For the same reason, a bent-tube boiler costs less when it is narrow and high. Since the manner in which boiler costs and efficiency vary with length, width, and height are known only by the boiler manufacturer, their selection should be left as far as possible to him. By specifying the desired capacity in pounds of steam per hour and the efficiency, and asking for bids from several manufacturers, the owner is assured of obtaining the lowest cost arrangement of heating surface.

683. In Selecting a Water-tube Boiler, Consideration Should Be Given to the Draft Pressure Which Is Available in the Plant.—If the draft pressure which is produced by the existing stack is low and it is necessary to utilize this stack unmodified, then care should be exercised to select a boiler of such type that the existing draft pressure will be sufficient for its operation. Usually a return-tubular boiler can be operated by a stack which would not operate an equivalent water-tube boiler. Method of firing must be considered at the same time.

684. The suitable applications for the Vertical fire-tube boilers (see Secs. 61 to 66 for description) are those which require a small-capacity simple self-contained low first-cost unit. The features just listed are about the only ones in favor of boilers of this type. The vertical fire-tube boilers are uneconomical; in fact, it is impossible to design coal-fired boilers, of the small sizes in which these are made, which will have both low and first-cost and high economy. Vertical fire

tube boilers have economies of possibly only 50 to 70 per cent of those of return-tubular boilers. The vertical fire-tube boilers will evaporate about 5 lb. of water per pound of coal as against about 8 lb. of water per lb. of coal for the return-tubular boilers. Vertical fire-tube boilers may be purchased in capacities of from 5 to, say, 75 boiler hp.

NOTE.—Since the vertical fire-tube boilers are about the only self-contained small-capacity units obtainable at reasonable cost, they are widely used for hoisting-engines and tractors, and in cleaning and dying establishments, restaurants, and bakeries, where the steam is used direct in the industrial processes. In many localities it is legal to operate such boilers (provided a specified steam pressure is not exceeded) without a licensed engineer.

685. The proper policy to pursue in taking advantage of overload capacity of boilers should be governed by both economical considerations and by local conditions of load and available space. It is readily recognized that a boiler capable of supplying a given load at a low rating will cost more to install than a boiler supplying the same load at a much higher rating. But the operating efficiency of the latter will be lower and maintenance higher. It therefore becomes a problem of balancing the higher fixed charges and lower operating costs of the larger boiler against the lower fixed charges and higher operating cost of the smaller boiler. It is obvious that duration of peak loads and "use factor" play a large part in any study of economical boiler size. Today complete boiler installations are usually guaranteed to be capable of generating so many thousand pounds of steam per hour continuously and a higher capacity for the duration of a 2-hr. peak.

NOTE.—It should be understood that overload capacity of a boiler ultimately reverts to a problem of proper furnace and stoker design. With an improperly designed furnace and an illogically selected stoker, even the very best boiler could not be made to carry overloads economically.

NOTE.—Where a boiler plant will operate 24 hr. daily, seven days each week, it is imperative that at least two boiler units be installed so that one of them may be shut down for cleaning while the other is being operated. But where the plant is small and can be shut down on Sundays, then, in certain cases, one boiler unit may suffice.

686. When determining boiler capacities for a plant upon which the imposed load is unsteady, first plot a load graph

using *pounds of steam per hour* for ordinates and *hours of the day* as abscissæ. (Examples of load graphs for loads of different characteristics are reproduced in the author's "Central Stations.") Such a graph will assist one in visualizing conditions and be of great service in selecting capacities such that the normal loads and overloads can be handled effectively.

687. Some general rules which may usually be followed in selecting boiler sizes are these:

1. All boilers in a plant should preferably be of the same size and type to insure interchangeability of parts and uniformity of equipment and operating methods.
2. The sizes should be so selected that, if possible, such boilers as are working in the plant should always be working at or near their most economical loads.
3. In general, return-tubular boilers should not, for economic reasons, be larger than 1,750 nor smaller than say 750 sq. ft. of heating surface (see Sec. 678).
4. Water-tube boilers should not (see Sec. 678) in general, for economic reasons, be smaller than about 1,500 sq. ft.

QUESTIONS ON DIVISION 26

1. Into what two classifications may the selection of steam boilers be divided?
2. Name the chief factors which should be considered in selecting the type of boiler for a given service.
3. Outline the general method of procedure in determining the most economical type of boiler.
4. Upon what basis may the selection of a return-tubular vs. a water-tube boiler usually be determined?
5. For what conditions of installations are vertical water-tube boilers usually selected?
6. What characteristics should a water-tube boiler have if it is to be subjected to heavy overloads?
7. In what way may the overload capacity of boilers be utilized?
8. Explain the method of procedure which may be employed in determining boiler capacities for a varying-load plant.
9. Give four general rules which may be followed in selecting boiler sizes.

APPENDIX

SOLUTIONS TO PROBLEMS

The following solutions to the problems, which have been presented at the ends of the various divisions throughout the book, are included to assist the student. These solutions should be referred to only after the reader has made an earnest effort to solve, without assistance, the problem which is under consideration. If used in this way, these solutions may constitute a material aid. But if the reader refers to this appendix before he has made an honest effort to work out his own solution, then, the material in this appendix will probably do more harm than good.

The same symbols and the same formulas are used in these solutions as those which are employed in the Division which precedes the problems which are proposed in the text portions of the book.

SOLUTIONS TO PROBLEMS ON DIVISION 9

STRESSES IN AND STRENGTHS OF STEAM BOILERS

1. $P_T = dLP_{gt} = (4 \times 12) \times (20 \times 12) \times 125 = 1{,}440{,}000$ lb.

2. $d = (P_T)/(LP_{gt}) = 1{,}152{,}000 \div [(24 \times 12) \times 120] = 1{,}152{,}000 \div 34{,}560 = 33.3$ in.

3. $P_T = rLP_{gt} = 21 \times (40 \times 12) \times 90 = 907{,}200$ lb.

4. $S_{Tt} = (2tU_tL)/f = 2 \times 0.375 \times 54{,}000 \times (20 \times 12) \div 5 = 1{,}944{,}000$ lb.

5. $P_{gt} = (2tU_t)/(df) = (2 \times 0.25 \times 45{,}000) \div (36 \times 4.5) = 2{,}500 \div 18 = 139$ lb. per sq. in.

6. $P_L = 0.7854d^2P_{gl} = 0.7854 \times 21^2 \times 150 = 51{,}954$ lb.

7. $S_{tl} = (3.14dtU_t)/f = (3.14 \times 21 \times 0.3125 \times 50{,}000) \div 5 = 206{,}060$ lb.

8. $P_{gl} = (4tU_t)/(df) = (4 \times 0.3125 \times 50{,}000) \div (21 \times 5) = (0.3125 \times 40{,}000) \div 21 = 12{,}500 \div 21 = 595$ lb. per sq. in.

9. One-half of roundabout $= 595 \div 2 = 297.5$ lb. per sq. in.

10. $P_{gl} = (4tU_t)/(df) = (4 \times 0.125 \times 45{,}000) \div (15 \times 5) = 0.500 \times 600 = 300$ lb. per sq. in.

11. $P_{MAW} = (TS \times t \times E)/(R \times FS) = (50{,}000 \times 0.375 \times 0.80) \div (18 \times 5) = 3{,}000 \div 18 = 166.7$ lb. per sq. in.

12. $t = (P_{MAW} \times R \times FS)/(TS \times E) = (200 \times 18 \times 5) \div (50{,}000 \times 0.80) = 18 \div 40 = 0.45$ in. or a little over $\frac{7}{16}$ in.

SOLUTIONS TO PROBLEMS ON DIVISION 10

RIVETED JOINTS

1. Strength of unit strip: $S_T = L_U t S_T = 2\frac{1}{2} \times \frac{5}{16} \times 55{,}000 = 42{,}969$ lb.

Strength of riveted unit strip: $S_T' = (L_U - d)tS_t = (2\frac{1}{2} - \frac{3}{4}) \times \frac{5}{16} \times 55{,}000 = 30{,}078$ lb.

Strength of rivets in shear: $S_s = NAS_s = 2 \times 0.442 \times 44{,}000 = 38{,}896$ lb.

Strength of plate against crushing: $S_c = NdtS_g = 2 \times \frac{3}{4} \times \frac{5}{16} \times 95{,}000 = 44{,}531$ lb.

Least strength: $S_T' = 30{,}078$ lb.

Therefore, efficiency $= \dfrac{S_T'}{S_T} = \dfrac{30{,}078}{42{,}969} = 0.70 = 70$ per cent.

SOLUTIONS TO PROBLEMS ON DIVISION 20

DRAFT AND ITS PRODUCTION AND MEASUREMENT

1.
$$P'_D = 0.52 L_h P_2 \left(\frac{1}{T_0 + 460} - \frac{1}{T_G + 460} \right)$$
$$P'_D = 0.52 \times 110 \times 14.7 \times \left(\frac{1}{65 + 460} - \frac{1}{550 + 460} \right)$$
$$P'_D = 840.8 \times \left(\frac{1}{525} - \frac{1}{1010} \right) = 840.8 \times (0.001{,}905 - 0.000{,}990)$$
$$= 840.8 \times 0.000{,}915 = 0.765$$
$P'_D = 0.77$-in. water column = total draft pressure.

2.
$$L_h = \frac{P'_D}{0.52 P_2 \left(\frac{1}{T_o + 460} - \frac{1}{T_G + 460} \right)}$$
$$L_h = \frac{2}{0.52 \times 13 \times \left(\frac{1}{55 + 460} - \frac{1}{500 + 460} \right)}$$
$$L_h = \frac{2}{6.76 \times \left(\frac{1}{515} - \frac{1}{960} \right)}$$
$$L_h = \frac{2}{6.76 \times (0.001{,}943 - 0.001{,}042)} = \frac{2}{6.76 \times 0.000{,}901}$$
$$L_h = \frac{2}{0.00609} = 328 \text{ ft.} = \text{height of chimney.}$$

3. Available draft = (elevation draft pressure) − (fire and velocity drop).

Available draft = 0.47 − 0.09.
Available draft = 0.38-in. water column.

4. $P''_D = 1.25(A.D.D._{BC})$.
$P''_D = 1.25 \times 0.75$.
$P''_D = 0.94$-in. water column = total draft pressure.

5. $A = \dfrac{W_c}{12\sqrt{L_h}}$

$$A = \frac{1.5 \times 2{,}000}{12\sqrt{120}} = \frac{3{,}000}{12 \times 10.9} = \frac{3{,}000}{131.5}$$

$A = 22.8$ sq. ft. = area of flue.

6. Coal burnt per hr. = 4 × 8 × 500 × 1.25 = 20,000 lb.
Total grate area = 83 × 8 = 664 sq. ft.
Lb. coal burnt per sq. ft. of grate surface per hr. = 20,000/664 = 30.12.
Hence (from Fig. 365) for Illinois bituminous coal, the available draft required for furnace and fuel bed is............. 0.34-in. water column
Draft required for breeching (see Table XVIII for values) is 50 × 0.001 =.................. 0.05-in. water column
Draft required for breeching elbows (see Table XVIII) is 2 × 0.05 =........................ 0.10-in. water column
Draft required for passes, assumed (Table XVIII) to be................................ 0.40-in. water column
Total available or effective draft required = $A.D.D._{BC}$.................................. 0.89-in. water column
Now, to find the total draft which the chimney must develop from the smoke-conduit connection up, substitute in formula (61):

$$P''_D = 1.25(A.D.D._{BC}) = 1.25 \times 0.89 = 1.11\text{-in. water column.}$$

Then to find the height of stack required to develop $P''_D = 1.11$-in. water column, substitute in formula (59):

$$L_h = \frac{P'_d}{0.52P_1\left(\frac{1}{T_o + 460} - \frac{1}{T_G + 460}\right)} = \frac{1.11}{0.52 \times 13.57 \times \left(\frac{1}{60 + 460} - \frac{1}{550 + 460}\right)}$$

$$L_h = \frac{1.11}{7.05 \times \left(\frac{1}{520} - \frac{1}{1{,}010}\right)} = \frac{1.11}{7.05 \times (0.001{,}925 - 0.000{,}990)}$$

$$L_h = \frac{1.11}{7.05 \times 0.000{,}935} = \frac{1.11}{0.006{,}59}$$

$L_h = 168$ ft. = height of chimney above smoke-conduit connectio~

The flue area of the chimney should, formula (66), be:

$$A = \frac{W_c}{12\sqrt{L_h}} = \frac{20{,}000}{12\sqrt{168}} = \frac{20{,}000}{12 \times 12.98} = \frac{20{,}000}{15?}$$

$A = 128$ sq. ft. = flue area

The flue diameter should, formula (67), be:

$$d = \sqrt{\frac{W_c}{9.43\sqrt{L_h}}} = \sqrt{\frac{20{,}000}{9.43\sqrt{168.5}}} = \sqrt{\frac{20{,}000}{9.43 \times 12.9}}$$

$$d = \sqrt{\frac{20{,}000}{121.8}} = \sqrt{164.5}$$

$d = 12.8$ ft. $= 12$ ft. 10 in. = chimney diameter.

SOLUTIONS TO PROBLEMS ON DIVISION 21

CHIMNEYS, BREECHINGS

1. $P = kv^2 = 0.003 \times 80^2 = 19.2$ lb. per sq. ft.
and $P = kv^2 = 0.0035 \times 80^2 = 22.4$ lb. per sq. ft.

2. $F = \frac{L_{wb} + L_{wt}}{2} L_h P = \frac{9.5 + 8}{2} 120 \times 30 = 31{,}500$ lb.

3. $L_{hc} = \frac{L_{wb} + 2L_{wt}}{L_{wb} + L_{wt}} \times \frac{L_h}{3} = \frac{9.5 + (2 \times 8)}{9.5 + 8} \times \frac{120}{3} = 58.3$ ft.

4. $p''_c = \frac{FL_{hc}}{I \div c} = \frac{FL_{hc}}{0.118L^3} = \frac{25{,}000 \times (52 \times 12)}{(16 \times 12)^3 \times 0.118} = 18.74$ lb. per sq. in.

5. $p'_c = \frac{W_t}{A} = \frac{300 \times 2{,}000}{(16 \times 12)^2} = 16.3$ lb. per sq. in. or 1.17 tons per sq. ft.

$p_c = p'_c + p''_c = 18.74 + 16.3 = 35.04$ lb. per sq. in. or 2.52 tons per sq. ft

6. $x = F\frac{L_{hc}}{W} = 30{,}000\frac{45 \times 12}{600{,}000} = 27$ in.

7. $p'_c = \frac{W_t}{0.7854(d_o^2 - di^2)} = \frac{485 \times 2{,}000}{0.7854[(11 \times 12)^2 - (8\frac{1}{2} \times 12)^2]} =$ 176.5 lb. per sq. in.

8. $p''_c = \frac{FL_h}{I \div c} = \frac{FL_h}{0.7854[(r_o^4 - r_i^4) \div r_o]} =$

$$\frac{40{,}000 \times (60 \times 12)}{0.7854[(5\frac{1}{2} \times 12)^4 - (4\frac{1}{4} \times 12)^4] \div (5\frac{1}{2} \times 12)} = 199 \text{ lb. per sq. in.}$$

9. $p_c = p'_c + p''_c = 176.5 + 198.5 = 375.0$ lb. per sq. in.

10. $P = kv^2 = 0.003 \times 95^2 = 27.075$ lb. per sq. in.
$F = A \times P = 11 \times 175 \times 27.075 = 52{,}119$ lb.

$p'''_c = \frac{FL_{hc}}{0.8d_o^2t} = \frac{52{,}119 \times (87.5 \times 12)}{0.8(11 \times 12)^2 \times 0.625} = 6{,}281$ lb. per sq. in.

Stack is safe.

11. Pull along guy $= \frac{\text{horizontal pull}}{\sin 55°} = \frac{18{,}000}{0.819} = 22{,}000$ lb. or 11 tons.

$\frac{7}{8}$-in. rope is required. Add 5,000 lb. for initial tension of $\frac{7}{8}$-in. guy. Total force along guy = 22,000 + 5,000 = 27,000 lb. Guy required is 1 in. diameter.

12. $t = 4 + 0.05d_i + 0.0005L_h = 4 + 0.05(132) + 0.0005(2{,}700) =$ 11.95 in. Practical thickness = $1\frac{1}{2}$ bricks + $\frac{1}{2}$-in. mortar joint = $12\frac{1}{2}$ in.

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